CHP

EPA COMBINED HEAT AND POWER PARTNERSHIP

Catalog of CHP Technologies

U.S. Environmental Protection Agency

Combined Heat and Power Partnership

Introduction to CHP Technologies

Introduction

Fueled by electric industry deregulation, environmental concerns, unease over energy security, and a host of other factors, interest in combined heat and power (CHP) technologies has been growing among energy customers, regulators, legislators, and developers. CHP is a specific form of distributed generation (DG), which refers to the strategic placement of electric power generating units at or near customer facilities to supply on-site energy needs. CHP enhances the advantages of DG by the simultaneous production of useful thermal and power output, thereby increasing the overall efficiency.

CHP offers energy and environmental benefits over electric-only and thermal-only systems in both central and distributed power generation applications. CHP systems have the potential for a wide range of applications and the higher efficiencies result in lower emissions than separate heat and power generation system. The advantages of CHP broadly include the following:

- The simultaneous production of useful thermal and electrical energy in CHP systems lead to increased fuel efficiency.
- CHP units can be strategically located at the point of energy use. Such onsite generation avoids the transmission and distribution losses associated with electricity purchased via the grid from central stations.
- CHP is versatile and can be coupled with existing and planned technologies for many different applications in the industrial, commercial, and residential sectors.

EPA offers this catalog of CHP technologies as an on-line educational resource for the regulatory, policy, permitting, and other communities. EPA recognizes that some energy projects will not be suitable for CHP; however, EPA hopes that this catalog will assist readers in identifying opportunities for CHP in applications where thermal-only or electric-only generation are currently being considered.

The remainder of this introductory summary is divided into sections. The first section provides a brief overview of how CHP systems work and the key concepts of efficiency and power-to-heat ratios. The second section summarizes the cost and performance characteristics of five CHP technologies in use and under development.

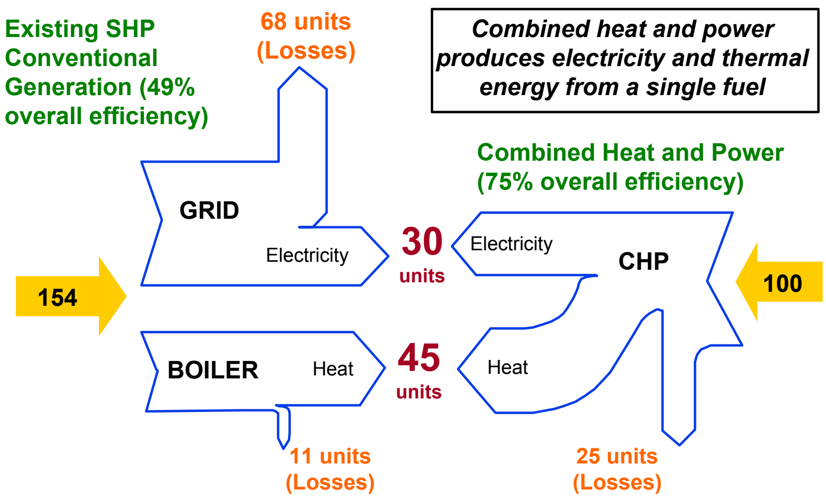
Overview of Combined Heat and Power

What is Combined Heat and Power?

CHP is the sequential or simultaneous generation of multiple forms of useful energy (usually mechanical and thermal) in a single, integrated system. CHP systems consist of a number of individual components – prime mover (heat engine), generator, heat recovery, and electrical interconnection – configured into an integrated whole. The type of equipment that drives the overall system (i.e., the prime mover) typically identifies the CHP system. Prime movers for CHP systems include reciprocating engines, combustion or gas turbines, steam turbines, microturbines, and fuel cells. These prime movers are capable of burning a variety of fuels, including natural gas, coal, oil, and alternative fuels to produce shaft power or mechanical energy. Although mechanical energy from the prime mover is most often used to drive a generator to produce electricity, it can also be used to drive rotating equipment such as compressors, pumps, and fans. Thermal energy from the system can be used in direct process applications or indirectly to produce steam, hot water, hot air for drying, or chilled water for process cooling.

Figure 1 shows the efficiency advantage of CHP compared with conventional central station power generation and on-site boilers. When considering both thermal and electrical processes together, CHP typically requires only ¾ the primary energy separate heat and power systems require. This reduced primary fuel consumption is key to the environmental benefits of CHP, since burning the same fuel more efficiently means fewer emissions for the same level of output.

Figure 1: CHP versus Separate Heat and Power (SHP) Production



Note: Assumes national averages for grid electricity and incorporates electric transmission losses.

Source: Tina Kaarsberg and Joseph Roop, "Combined Heat and Power: How Much Carbon and Energy Can It Save for Manufacturers?"

Expressing CHP Efficiency

Many of the benefits of CHP stem from the relatively high efficiency of CHP systems compared to other systems. Because CHP systems simultaneously produce electricity and useful thermal energy, CHP efficiency is measured and expressed in a number of different ways.[1] Table I summarizes the key elements of efficiency as applied to CHP systems.

[1] Measures of efficiency are denoted either as lower heating value (LHV) or higher heating value (HHV). HHV includes the heat of condensation of the water vapor in the products. Unless otherwise noted, all efficiency measures in this section are reported on an HHV basis.

Table I: Measuring the Efficiency of CHP Systems			
System	**Component**	**Efficiency Measure**	**Description**
Separate heat and power (SHP)	Thermal Efficiency (Boiler)	$EFF_Q = \frac{\text{Net Useful Thermal Output}}{\text{Energy Input}}$	Net useful thermal output for the fuel consumed
	Electric-only generation	$EFF_P = \frac{\text{Power Output}}{\text{Energy Input}}$	Electricity Purchased From Central Stations via Transmission Grid
	Overall Efficiency of separate heat and power (SHP)	$EFF_{SHP} = \frac{P + Q}{P/EFF_{Power} + Q/EFF_{Thermal}}$	Sum of net power (P) and useful thermal energy output (Q) divided by the sum of fuel consumed to produce each.
Combined heat and power (CHP)	Total CHP System Efficiency	$EFF_{Total} = (P + Q)/F$	Sum of the net power and net useful thermal output divided by the total fuel (F) consumed.
	FERC Efficiency Standard	$EFF_{FERC} = \frac{(P + Q/2)}{F}$	Developed for the Public Utilities Regulatory Act of 1978, the FERC methodology attempts to recognize the quality of electrical output relative to thermal output.
	Effective Electrical Efficiency (or Fuel Utilization Efficiency, FUE):	$FUE = \frac{P}{F - Q/EFF_{Thermal}}$	Ratio of net power output to net fuel consumption, where net fuel consumption excludes the portion of fuel used for producing useful heat output. Fuel used to produce useful heat is calculated assuming typical boiler efficiency, usually 80%.
	Percent Fuel Savings	$S = 1 - \frac{F}{P/EFF_P + Q/EFF_Q}$	Fuel savings compares the fuel used by the CHP system to a separate heat and power system. Positive values represent fuel savings while negative values indicate that the CHP system is using more fuel than SHP.

Key:
P = Net power output from CHP system
Q = Net useful thermal energy from CHP system
F = Total fuel input to CHP system
EFF_P = Efficiency of displaced electric generation
EFF_Q = Efficiency of displaced thermal generation

As illustrated in Table I the efficiency of electricity generation in power-only systems is determined by the relationship between net electrical output and the amount of fuel used for the power generation. **Heat rate**, the term often used to express efficiency in such power generation systems, is represented in terms of Btus of fuel consumed per kWh of electricity generated. However, CHP plants produce useable heat as well as electricity. In CHP systems, the **total CHP efficiency** seeks to capture the energy content of both electricity and usable steam and is the net electrical output plus the net useful thermal output of the CHP system divided by the fuel consumed in the production of electricity and steam. While total CHP efficiency provides a measure for capturing the energy content of electricity and steam produced it does not adequately reflect the fact that electricity and steam have different qualities. The quality and value of electrical output is higher relative to heat output and is evidenced by the fact that electricity can be transmitted over long distances and can be converted to other forms of energy. To account for these differences in quality, the Public Utilities Regulatory Policies Act of 1978 (PURPA) discounts half of the thermal energy in its calculation of the efficiency standard (Eff_{FERC}). The EFF_{FERC} is represented as the ratio of net electric output plus half of the net thermal output to the total fuel used in the CHP system. Opinions vary as to whether the standard was arbitrarily set, but the FERC methodology does recognize the value of different forms of energy. The following equation calculates the FERC efficiency value for CHP applications.

$$EFF_{FERC} = \frac{P + Q/2}{F}$$

Where: P = Net power output from CHP system
F = Total fuel input to CHP system
Q = Net thermal energy from CHP system

Another definition of CHP efficiency is **effective electrical efficiency**, also known as **fuel utilization effectiveness (FUE)**. This measure expresses CHP efficiency as the ratio of net electrical output to net fuel consumption, where net fuel consumption excludes the portion of fuel that goes to producing useful heat output. The fuel used to produce useful heat is calculated assuming typical boiler efficiency, generally 80%. The effective electrical efficiency measure for CHP captures the value of both the electrical and thermal outputs of CHP plants. The following equation calculates FEU.

$$FUE = \frac{P}{F - Q/EFF_Q}$$

Where: Eff_Q = Efficiency of displaced thermal generation

FEU captures the value of both the electrical and thermal outputs of CHP plants and it specifically measures the efficiency of generating power through the incremental fuel consumption of the CHP system.

EPA considers fuel savings as the appropriate term to use when discussing CHP benefits relative to separate heat and power (SHP) operations. Fuel savings compares the fuel used by the CHP system to a separate heat and power system (i.e. boiler and electric-only generation). The following equation determines percent fuel savings (S).

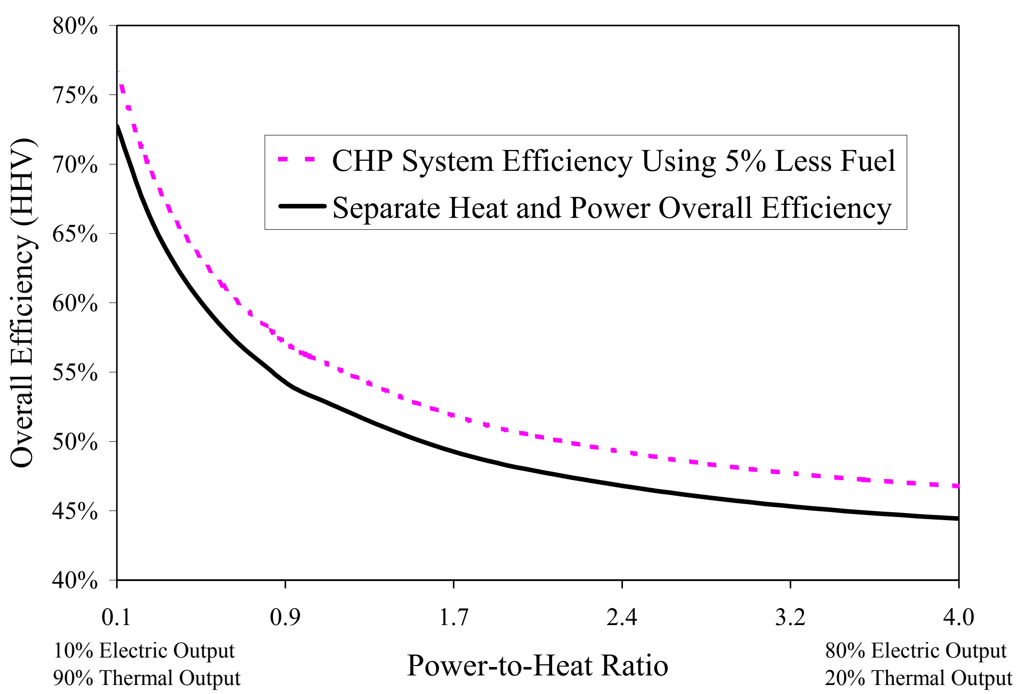
$$S = 1 - \left[\frac{F}{P/Eff_P + Q/Eff_Q} \right]$$

Where:
Eff_P = Efficiency of displaced electric generation
Eff_Q = Efficiency of displaced thermal-only facility

In the fuel saving equation given above, the numerator in the bracket term denotes the fuel used in the production of electricity and steam in a CHP system. The denominator describes the sum of the fuel used in the production of electricity (P/Eff_P) and thermal energy (Q/Eff_Q) in separate heat-and-power operations. Positive values represent fuel savings while negative values indicate that the CHP system in question is using more fuel than separate heat and power generation.

Another important concept related to CHP efficiency is the **power-to-heat ratio**. The power-to-heat ratio indicates the proportion of power (electrical or mechanical energy) to heat energy (steam or hot water) produced in the CHP system. Because the efficiencies of power generation and steam generation are likely to be considerably different, the power-to-heat ratio has an important bearing on how the total CHP system efficiency of the CHP system might compare to a separate power-and-heat system. Figure 2 illustrates this point. The illustrative curves display how the overall efficiency might change under alternate power-to-heat ratios for a separate power-and-heat system and a CHP system (for illustrative purposes, the CHP system is assumed to use 5% less fuel than its separate heat-and-power counterpart for the same level of electrical and thermal output).

Figure 2: Equivalent Separate Heat and Power Efficiency
Assumes 40% efficient electric and 80% efficient thermal generation



Overview of CHP Technologies

This catalog is comprised of five chapters that characterize each of the different CHP technologies (gas turbine, reciprocating engines, steam turbines, microturbines, and fuel cells) in detail. Many of these technologies are commonly used today, some are in the early stages of commercialization, and others are expected to be available in a few years. The chapters supply information on the applications of the technology, detailed descriptions of its functionality and design characteristics, performance characteristics, emissions, and emissions control options. The following sections provide snapshots of the five technologies, and a comparison of key cost and performance characteristics across the range of technologies that highlights the distinctiveness of each. Tables II and III provide a summary of the key cost and performance characteristics of the CHP technologies discussed in the catalog.

<table>
<tr><th colspan="4">Table II: Summary of CHP Technologies</th></tr>
<tr><th>CHP system</th><th>Advantages</th><th>Disadvantages</th><th>Available sizes</th></tr>
<tr><td>Gas turbine</td><td>High reliability.
Low emissions.
High grade heat available.
No cooling required.</td><td>Require high pressure gas or in-house gas compressor.
Poor efficiency at low loading.
Output falls as ambient temperature rises.</td><td>500 kW to 40 MW</td></tr>
<tr><td>Microturbine</td><td>Small number of moving parts.
Compact size and light weight.
Low emissions.
No cooling required.</td><td>High costs.
Relatively low mechanical efficiency.
Limited to lower temperature cogeneration applications.</td><td>30 kW to 350 kW</td></tr>
<tr><td>Spark ignition (SI) reciprocating engine</td><td rowspan="3">High power efficiency with part-load operational flexibility.
Fast start-up.
Relatively low investment cost.
Can be used in island mode and have good load following capability.
Can be overhauled on site with normal operators.
Operate on low-pressure gas.</td><td rowspan="3">High maintenance costs.
Limited to lower temperature cogeneration applications.
Relatively high air emissions.
Must be cooled even if recovered heat is not used.
High levels of low frequency noise.</td><td>< 5 MW</td></tr>
<tr><td rowspan="2">Diesel/compression ignition (CI) reciprocating engine</td><td>High speed (1,200 RPM) ≤4MW</td></tr>
<tr><td>Low speed (60-275 RPM) ≤65MW</td></tr>
<tr><td>Steam turbine</td><td>High overall efficiency.
Any type of fuel may be used.
Ability to meet more than one site heat grade requirement.
Long working life and high reliability.
Power to heat ratio can be varied.</td><td>Slow start up.
Low power to heat ratio.</td><td>50 kW to 250 MW</td></tr>
<tr><td>Fuel Cells</td><td>Low emissions and low noise.
High efficiency over load range.
Modular design.</td><td>High costs.
Low durability and power density.
Fuels requiring processing unless pure hydrogen is used.</td><td>200 kW to 250 kW</td></tr>
</table>

Table III: Summary Table of Typical Cost and Performance Characteristics by CHP Technology Type*						
Technology	**Steam turbine[1]**	**Diesel engine**	**Nat. gas engine**	**Gas turbine**	**Microturbine**	**Fuel cell**
Power efficiency (HHV)	15-38%	27-45%	22-40%	22-36%	18-27%	30-63%
Overall efficiency (HHV)	80%	70-80%	70-80%	70-75%	65-75%	65-80%
Effective electrical efficiency	75%	70-80%	70-80%	50-70%	50-70%	60-80%
Typical capacity (MW_e)	0.2-800	0.03-5	0.05-5	1-500	0.03-0.35	0.01-2
Typical power to heat ratio	0.1-0.3	0.5-1	0.5-1	0.5-2	0.4-0.7	1-2
Part-load	ok	good	ok	poor	ok	good
CHP Installed costs ($/$kW_e$)	300-900	900-1,500	900-1,500	800-1,800	1,300-2,500	2,700-5,300
O&M costs ($/$kWh_e$)	<0.004	0.005-0.015	0.007-0.02	0.003-0.0096	0.01 (projected)	0.005-0.04
Availability	near 100%	90-95%	92-97%	90-98%	90-98%	>95%
Hours to overhauls	>50,000	25,000-30,000	24,000-60,000	30,000-50,000	5,000-40,000	10,000-40,000
Start-up time	1 hr - 1 day	10 sec	10 sec	10 min - 1 hr	60 sec	3 hrs - 2 days
Fuel pressure (psi)	n/a	<5	1-45	120-500 (compressor)	40-100 (compresor)	0.5-45
Fuels	all	diesel, residual oil	natural gas, biogas, propane, landfill gas	natural gas, biogas, propane, oil	natural gas, biogas, propane, oil	hydrogen, natural gas, propane, methanol
Noise	high	high	high	moderate	moderate	low
Uses for thermal output	LP-HP steam	hot water, LP steam	hot water, LP steam	heat, hot water, LP-HP steam	heat, hot water, LP steam	hot water, LP-HP steam
Power Density (kW/m^2)	>100	35-50	35-50	20-500	5-70	5-20
NO_x[2], lb/MMbtu	0.03-0.3	1-1.8[3]	0.18	0.05	0.03	0.004
$lb/MWh_{TotalOutput}$	0.13-1.3	4.3-8.2[4]	0.8	0.25	0.15	0.02

* Data are illustrative values for 'typically' available systems; All $ are in 2000$

[1] For steam turbine, not entire boiler package

[2] New low emitting units without end of pipe controls

[3] Present on road diesel requirements are approximately 1 lb/MMBtu, but most backup diesel generators emit at 1.8 lb/MMBtu

[4] New on road diesel rule would bring emissions rate to approximately 0.3 $lb/MWh_{TotalOutput}$

Technology

The first chapter of the catalog focuses on gas turbines as a CHP technology. Gas turbines are typically available in sizes ranging from 500 kW to 250 MW and can operate on a variety of fuels such as natural gas, synthetic gas, landfill gas, and fuel oils. Most gas turbines typically operate on gaseous fuel with liquid fuel as a back up. Gas turbines can be used in a variety of configurations including (1) simple cycle operation with a single gas turbine producing power only, (2) combined heat and power (CHP) operation with a single gas turbine coupled and a heat recovery exchanger and (3) combined cycle operation in which high pressure steam is generated from recovered exhaust heat and used to produce additional power using a steam turbine. Some combined cycles systems extract steam at an intermediate pressure for use and are combined cycle CHP systems. Many industrial and institutional facilities have successfully used gas turbines in CHP mode to generate power and thermal energy on-site. Gas turbines are well suited for CHP because their high-temperature exhaust can be used to generate process steam at conditions as high as 1,200 pounds per square inch gauge (psig) and 900 degree Fahrenheit (ºF). Much of the gas turbine-based CHP capacity currently existing in the United States consists of large combined-cycle CHP systems that maximize power production for sale to the grid. Simple-cycle CHP applications are common in smaller installations, typically less than 40 MW.

The second chapter of the catalog focuses on microturbines, which are small electricity generators that can burn a wide variety of fuels including natural gas, sour gases (high sulfur, low Btu content), and liquid fuels such as gasoline, kerosene, and diesel fuel/distillate heating oil. Microturbines use the fuel to create high-speed rotation that turns an electrical generator to produce electricity. In CHP operation, a heat exchanger referred to as the exhaust gas heat exchanger, transfers thermal energy from the microturbine exhaust to a hot water system. Exhaust heat can be used for a number of different applications including potable water heating, absorption chillers and desiccant dehumidification equipment, space heating, process heating, and other building uses. Microturbines entered field-testing in 1997 and the first units began commercial service in 2000. Available and models under development typically range in sizes from 30 kW to 350 kW.

The third chapter in the catalog describes the various types of reciprocating engines used in CHP applications. Spark ignition (SI) and compression ignition (CI) are the most common types of reciprocating engines used in CHP-related projects. SI engines use spark plugs with a high-intensity spark of timed duration to ignite a compressed fuel-air mixture within the cylinder. SI engines are available in sizes up to 5 MW. Natural gas is the preferred fuel in electric generation and CHP applications of SI; however, propane, gasoline and landfill gas can also be used. Diesel engines, also called CI engines, are among the most efficient simple-cycle power generation options in the market. These engines operate on diesel fuel or heavy oil. Dual fuel engines, which are diesel compression ignition engines predominantly fueled by natural gas with a small amount of diesel pilot fuel, are also used. Higher speed diesel engines (1,200 rpm) are available up to 4 MW in size, while lower speed diesel engines (60 - 275 rpm) can be as large as 65 MW. Reciprocating engines start quickly, follow load well, have good part-load efficiencies, and generally have high reliabilities. In many instances, multiple reciprocating engine units can be used to enhance plant capacity and availability. Reciprocating engines are well suited for applications that require hot water or low-pressure steam.

The fourth chapter of the catalog is dedicated to steam turbines that generate electricity from the heat (steam) produced in a boiler. The energy produced in the boiler is transferred to the turbine through high-pressure steam that in turn powers the turbine and generator. This separation of functions enables steam turbines to operate with a variety of fuels including natural gas, solid waste, coal, wood, wood waste, and agricultural by-products. The capacity of commercially available steam turbine typically ranges between 50 kW to over 250 MW. Although steam turbines are competitively priced compared to other prime movers, the costs of a complete boiler/steam turbine CHP system is relatively high on a per kW basis. This is because steam turbines are typically sized with low power to heat (P/H) ratios, and have high capital costs associated with the fuel and steam handling systems and the custom nature of most installations. Thus the ideal applications of steam turbine-based CHP systems include medium- and large-scale industrial or institutional facilities with high thermal loads and where solid or waste fuels are readily available for boiler use.

Chapter five in the catalog deals with an emerging technology that has the potential to serve power and thermal needs cleanly and efficiently. Fuel cells use an electrochemical or battery-like process to convert the chemical energy of hydrogen into water and electricity. In CHP applications, heat is generally recovered in the form of hot water or low-pressure steam (<30 psig) and the quality of heat is dependent on the type of fuel cell and its operating temperature. Fuel cells use hydrogen, which can be obtained from natural gas, coal gas, methanol, and other hydrocarbon fuels. There are currently five types of fuel cells under development. These include (1) phosphoric acid (PAFC), (2) proton exchange membrane (PEMFC), (3) molten carbonate (MCFC), (4) solid oxide (SOFC), and (5) alkaline (AFC). Currently, there are only two commercially available fuel cells, a 200 kW PAFC unit and a 250 kW MCFC unit. Due to the high installed cost of fuel cell systems, the most prominent DG applications of fuel cell systems are CHP-related.

Installed cost[2]

The total plant cost or installed cost for most CHP technologies consists of the total equipment cost plus installation labor and materials, engineering, project management, and financial carrying costs during the construction period. The cost of the basic technology package plus the costs for added systems needed for the particular application comprise the total equipment cost.

Total installed costs for gas turbines, microturbines, reciprocating engines, and steam turbines are comparable. The total installed cost for typical gas turbines ranges from $785/kW to $1,780/kW while total installed costs for typical microturbines in grid-interconnected CHP applications may range anywhere from $1,339/kW to $2,516/kW. Commercially available natural gas spark-ignited engine gensets have total installed costs of $920/kW to $1,515/kW, and steam turbines have total installed costs ranging from $349/kW to $918/kW. Fuel cells are currently the most expensive among the five CHP technologies with total installed costs ranging between $4,500/kW (for a 200 PAFC unit) to $5,000/kW (for a 250 MCFC unit).

O&M Cost

Non-fuel operation and maintenance (O&M) costs typically include routine inspections, scheduled overhauls, preventive maintenance, and operating labor. O&M costs are comparable for gas turbines, gas engine gensets, steam turbines and fuel cells, and only a fraction higher for microturbines. Total O&M costs range from $4.2/MWh to $9.6/MWh for typical gas turbines, from $9.3/MWh to $18.4/MWh for commercially available gas engine gensets and are typically

[2] All $ are 2000$.

less than $4/MWh for steam turbines. Based on manufacturers offer service contracts for specialized maintenance, the O&M costs for microturbines appear to be around $10/MWh. For fuel cells O&M costs range approximately between $29/MW and $43/MW.

Start-up time

Start-up times for the five CHP technologies described in this catalog can vary significantly depending on the technology and fuel used. Gas turbines have relatively short start up time, though heat recovery considerations may constraint start up times. Microturbines require several minutes for start-up but requires a power storage unit (typically a battery UPS) for start-up if the microturbine system is operating independent of the grid. Reciprocating engines have fast start-up capability, which allows for timely resumption of the system following a maintenance procedure. In peaking or emergency power applications, reciprocating engines can most quickly supply electricity on demand. Steam turbines, on the other hand, require long warm-up periods in order to obtain reliable service and prevent excessive thermal expansion, stress and wear. Fuel cells also have relatively long start-up times (especially for MCFC and SOFC). The longer start-up times for steam turbines and fuel cells make it more applicable to baseload needs.

Availability

Availability indicates the amount of time a unit can be used for electricity and/or steam production. Availability generally depends on the operational conditions of the unit. Frequent starts and stops of gas turbines can increase the likelihood of mechanical failure, though steady operation with clean fuels can permit gas turbines to operate for about a year without a shutdown. The estimated availability for gas turbines operating on clean gaseous fuels such as natural gas is over 95 percent.

Given the limited number of microturbines currently in commercial use it is difficult to draw conclusions on the reliability and availability of these units. At the same time, the basic design and limited number of moving parts in microturbines suggests that the technology will have good availability. Manufacturers of microturbines have targeted availabilities between 98 and 99 percent. Natural gas engine availabilities generally vary with engine type, speed, and fuel quality. Typically demonstrated availabilities for natural gas engine gensets in CHP applications is approximately 95 percent. Steam turbines have high availability rates -- usually greater than 99 percent with longer than one year between shutdowns for maintenance and inspections. However, for purposes of CHP application it should be noted that this high availability rate is only applicable to the steam turbine itself and not to the boiler or HRSG that is supplying the steam. Some demonstrated and commercially available fuel cells have achieved greater than 90 percent availability.

Thermal output

The ability to produce useful thermal energy from exhaust gases is the primary advantage of CHP technologies. Gas turbines produce a high quality (high temperature) thermal output suitable for most CHP applications. High-pressure steam can be generated or the exhaust can be used directly for process heating and drying. Microturbines produce exhaust output at temperatures in the 400ºF - 600ºF range, suitable for supplying a variety of building thermal needs. Reciprocating engines can produce hot water and low-pressure steam. Steam turbines are capable of operating over a broad range of steam pressures. They are custom designed to deliver the thermal requirements of CHP applications through use of backpressure or extraction steam at the appropriately needed pressure and temperature. Waste heat from fuel cells can be used primarily for domestic hot water and space heating applications.

Efficiency

Total CHP efficiency is a composite measure of the CHP fuel conversion capability and is expressed as the ratio of net output to fuel consumed. As explained earlier, for any technology the total CHP efficiency will vary depending on size and power-to-heat ratio. Combustion turbines achieve higher efficiencies at greater size and with higher power-to-heat ratios. The total CHP efficiency for gas turbines between 1 MW and 40 MW range from 70 percent to 75 percent for power-to-heat ratio between 0.5 to 1.0 respectively. Unlike gas turbines, microturbines typically achieve 65 percent to 75 percent total CHP efficiency for a range of power-to-heat ratios. Commercially available natural gas spark engines ranging between 100 kW to 5 MW are likely to have total CHP efficiency in the 75 percent to 80 percent. The total CHP efficiency of such engines will decrease with unit-size, and also with higher power-to-heat ratios. Although performance of steam turbines may differ substantially based on the fuel used, they are likely to achieve near 80 percent total CHP efficiency across a range of sizes and power-to–heat ratios. Fuel cell technologies may achieve total CHP efficiency in the 65 percent to 75 percent range.

Emissions

In addition to cost savings, CHP technologies offer significantly lower emissions rates compared to separate heat and power systems. The primary pollutants from gas turbines are oxides of nitrogen (NO_x), carbon monoxide (CO), and volatile organic compounds (VOCs) (unburned, non-methane hydrocarbons). Other pollutants such as oxides of sulfur (SO_x) and particulate matter (PM) are primarily dependent on the fuel used. Similarly emissions of carbon dioxide are also dependent on the fuel used. Many gas turbines burning gaseous fuels (mainly natural gas) feature lean premixed burners (also called dry low-NO_x burners) that produce NO_x emissions ranging between 0.3 lbs/MWh to 2.5 lbs/MWh with no post combustion emissions control. Typically commercially available gas turbines have CO emissions rates ranging between 0.4 lbs/MWh – 0.9 lbs/MWh. Selective catalytic reduction (SCR) or catalytic combustion can further help to reduce NO_x emissions by 80 percent to 90 percent from the gas turbine exhaust and carbon-monoxide oxidation catalysts can help to reduce CO by approximately 90 percent. Many gas turbines sited in locales with stringent emission regulations use SCR after-treatment to achieve extremely low NO_x emissions.

Microturbines have the potential for low emissions. All microturbines operating on gaseous fuels feature lean premixed (dry low NO_x, or DLN) combustor technology. The primary pollutants from microturbines include NO_x, CO, and unburned hydrocarbons. They also produce a negligible amount of SO_2. Microturbines are designed to achieve low emissions at full load and emissions are often higher when operating at part load. Typical NO_x emissions for microturbine systems range between 0.5 lbs/MWh and 0.8 lbs/MWh. Additional NO_x emissions removal from catalytic combustion is microturbines is unlikely to be pursued in the near term because of the dry low NO_x technology and the low turbine inlet temperature. CO emissions rates for microturbines typically range between 0.3 lbs/MWh and 1.5 lbs/MWh.
Exhaust emissions are the primary environmental concern with reciprocating engines. The primary pollutants from reciprocating engines are NO_x, CO, and VOCs. Other pollutants such as SO_x and PM are primarily dependent on the fuel used. The sulfur content of the fuel determines emissions of sulfur compounds, primarily SO_2. NO_x emissions from reciprocating engines typically range between 1.5 lbs/MWh to 44 lbs/MWh without any exhaust treatment.

Use of an oxidation catalyst or a three way conversion process (non-selective catalytic reductions) could help to lower the emissions of NO_x, CO and VOCs by 80 percent to 90 percent. Lean burn engines also achieve lower emissions rates than rich burn engines.

Emissions from steam turbines depend on the fuel used in the boiler or other steam sources, boiler furnace combustion section design, operation, and exhaust cleanup systems. Boiler emissions include NO_x, SO_x, PM, and CO. The emissions rates in steam turbine depend largely on the type of fuel used in the boiler. Typical boiler emissions rates for NO_x with any post-combustion treatment range between 0.2 lbs/MWh and 1.24 lbs/MMBtu for coal, 0.22 lbs/MMBtu to 0.49 lbs/MMBtu for wood, 0.15 lbs/MMBtu to 0.37 lbs/MMBtu for fuel oil, and 0.03lbs/MMBtu – 0.28 lbs/MMBtu for natural gas. Uncontrolled CO emissions rates range between 0.02 lbs/MMBtu to 0.7 lbs/MMBtu for coal, approximately 0.06 lbs/MMBtu for wood, 0.03 lbs/MMBtu for fuel oil and 0.08 lbs/MMBtu for natural gas. A variety of commercially available combustion and post-combustion NO_x reduction techniques exist with selective catalytic reductions achieving reductions as high as 90 percent.

SO_2 emissions from steam turbine depend largely on the sulfur content of the fuel used in the combustion process. SO_2 composes about 95% of the emitted sulfur and the remaining 5 percent are emitted as sulfur tri-oxide (SO_3). Flue gas desulphurization (FGD) is the most commonly used post-combustion SO_2 removal technology and is applicable to a broad range of different uses. FGD can provide up to 95 percent SO_2 removal.

Fuel cell systems have low emissions profiles because the primary power generation process does not involve combustion. The fuel processing subsystem is the only significant source of emissions as it converts fuel into hydrogen and low energy hydrogen exhaust stream. The hydrogen exhaust stream is combusted in the fuel processor to provide heat, achieving emissions signatures of less than 0.07 lbs/MWh of CO, less than 0.06 lbs/MWh of NO_x and negligible SO_x without any after-treatment for emissions. Fuel cells are not expected to require any emissions control devices to meet current and projected regulations.

While not considered a pollutant in the ordinary sense of directly affecting health, CO_2 emissions do result from the use the fossil fuel based CHP technologies. The amount of CO2 emitted in any of the CHP technologies discussed above depends on the fuel carbon content and the system efficiency. The fuel carbon content of natural gas is 34 lbs carbon/MMBtu; oil is 48 lbs of carbon/MMBtu and ash-free coal is 66 lbs of carbon/MMBtu.

Appendix 1: Fuel Savings Equations

<u>Absolute Fuel Savings:</u>

$$F_{CHP} = F_{SHP} * (1 - S) \text{ and } E_{SHP} = E_{CHP} * (1\text{-}S)$$

$$\text{Fuel Savings} = F_{SHP} - F_{CHP} = \frac{F_{CHP}}{1 - S} - F_{CHP}$$

Where F_{CHP} = CHP fuel use
F_{SHP} = SHP fuel use
S = % fuel savings compared to SHP
E_{CHP} = CHP efficiency
E_{SHP} = SHP efficiency

$$= F_{CHP}\left[\frac{1}{1-S} - 1\right] = F_{CHP}\left[\frac{1}{1-S} - \frac{1-S}{1-S}\right] = F_{CHP}\left[\frac{1-1+S}{1-S}\right]$$

$$\text{Fuel Savings} = F_{CHP}\left[\frac{S}{1-S}\right] = F_{SHP} - F_{SHP} * (1 - S) = F_{SHP} * S$$

<u>Percentage Fuel Savings</u>:

Equivalent separate heat and power (SHP) efficiency

$$Eff_{SHP} = \frac{\text{SHP Output}}{\text{SHP Fuel Input}} = \frac{P + Q}{P/Eff_P + Q/Eff_Q}$$

Where P = power output
Q = useful thermal output
Eff_P = power generation efficiency
Eff_Q = thermal generation efficiency

divide numerator and denominator by (P+Q)

$$Eff_{SHP} = \frac{1}{\frac{\%P}{Eff_P} + \frac{\%Q}{Eff_Q}}$$

Where %P = P/(P+Q)
%Q = Q/(P+Q)

CHP efficiency

$$Eff_{CHP} = \frac{P + Q}{F_{CHP}} = \frac{Eff_{SHP}}{(1 - S)}$$

Substitute in equation for EFF_{SHP} and isolate S

$$\frac{P + Q}{F} = \frac{\frac{P + Q}{P/EFF_P + Q/EFF_Q}}{(1 - S)}$$

$$(1-S)*\frac{P+Q}{F}=\frac{P+Q}{P/EFF_P+Q/EFF_Q}$$

Divide out (P+Q) and multiply by F

$$1-S=\frac{F}{\left(\frac{P}{Eff_P}+\frac{Q}{Eff_Q}\right)}$$

Percent fuel savings calculated from power and thermal output, CHP fuel input, and efficiency of displaced separate heat and power.

$$\boxed{S=1-\frac{F}{\frac{P}{Eff_P}+\frac{Q}{Eff_Q}}}$$

<u>Calculation of percentage power or percent thermal output from power to heat ratio:</u>

$$\text{Power to Heat Ratio}=X=P/Q=\%P/\%Q$$

$$P+Q=1$$

$$P=X*Q$$

$$P=X*(1-P)$$

$$P=X-X*P$$

$$P+X*P=X$$

$$P*(1+X)=X$$

$$P=\frac{X}{1+X}$$

$$Q=\frac{P}{X}$$

$$Q=\frac{1-Q}{X}$$

$$Q*X=1-Q$$

$$Q*(X+1)=1$$

$$Q=\frac{1}{X+1}$$

Technology Characterization: Gas Turbines

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Technology Characterization – Gas Turbines

Introduction and Summary

Engineering advancement pioneered the gas turbine in the early 1900s, and turbines began to be used for stationary electric power generation in the late 1930s. Turbines revolutionized airplane propulsion in the 1940s, and in the 1990s through today (early 2000s) are currently the economic and environmentally preferred choice for new power generation plants in the United States.

Gas turbines can be used in a variety of configurations: (1) simple cycle operation which is a single gas turbine producing power only, (2) combined heat and power (CHP) operation which is a simple cycle gas turbine with a heat recovery heat exchanger which recovers the heat in the turbine exhaust and converts it to useful thermal energy usually in the form of steam or hot water, ands (3) combined cycle operation in which high pressure steam is generated from recovered exhaust heat and used to create additional power using a steam turbine. Some combined cycles extract steam at an intermediate pressure for use in industrial processes and are combined cycle CHP systems.

Gas turbines are available in sizes ranging from 500 kilowatts (kW) to 250 megawatts (MW). The most efficient commercial technology for central station power-only generation is the gas turbine-steam turbine combined-cycle plant, with efficiencies approaching 60% (LHV).[1] Simple-cycle gas turbines for power-only generation are available with efficiencies approaching 40% (LHV). Gas turbines have long been used by utilities for peaking capacity. However, with changes in the power industry and advancements in the technology, the gas turbine is now being increasingly used for base-load power.

Gas turbines produce high-quality exhaust heat that can be used in CHP configurations to reach overall system efficiencies (electricity and useful thermal energy) of 70 to 80%. By the early 1980s, the efficiency and reliability of smaller gas turbines (1 to 40 MW) had progressed sufficiently to be an attractive choice for industrial and large institutional users for CHP applications.

Gas turbines are one of the cleanest means of generating electricity, with emissions of oxides of nitrogen (NO_x) from some large turbines in the single-digit parts per million (ppm) range, either with catalytic exhaust cleanup or lean pre-mixed combustion. Because of their relatively high efficiency and reliance on natural gas as the primary fuel, gas turbines emit substantially less

[1] Lower Heating Value. Most of the efficiencies quoted in this report are based on higher heating value (HHV), which includes the heat of condensation of the water vapor in the combustion products. In engineering and scientific literature the lower heating value (LHV – which does not include the heat of condensation of the water vapor in the combustion products) is often used. The HHV is greater than the LHV by approximately 10% with natural gas as the fuel (i.e., 50% LHV is equivalent to 45% HHV). HHV efficiencies are about 8% greater for oil (liquid petroleum products) and 5% for coal.

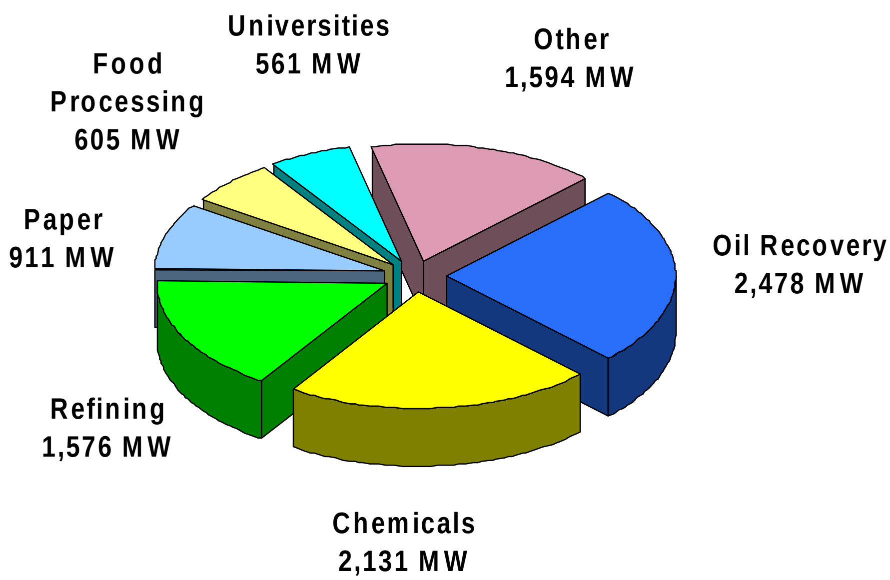
carbon dioxide (CO_2) per kilowatt-hour (kWh) generated than any other fossil technology in general commercial use.[2]

Applications

The oil and gas industry commonly use gas turbines to drive pumps and compressors, process industries use them to drive compressors and other large mechanical equipment, and many industrial and institutional facilities use turbines to generate electricity for use on-site. When used to generate power on-site, gas turbines are often used in combined heat and power mode where energy in the turbine exhaust provides thermal energy to the facility.

There were an estimated 40,000 MW of gas turbine based CHP capacity operating in the United States in 2000 located at over 575 industrial and institutional facilities.[3] Much of this capacity is concentrated in large combined-cycle CHP systems that maximize power production for sale to the grid. However, a significant number of simple-cycle gas turbine based CHP systems are in operation at a variety of applications as shown in **Figure 1**. Simple-cycle CHP applications are most prevalent in smaller installations, typically less than 40 MW.

Figure 1. Existing Simple Cycle Gas Turbine CHP – 9,854 MW at 359 sites



Source: PA Consulting; Energy Nexus Group

Gas turbines are ideally suited for CHP applications because their high-temperature exhaust can be used to generate process steam at conditions as high as 1,200 pounds per square inch gauge (psig) and 900 degree Fahrenheit (°F) or used directly in industrial processes for heating or drying. A typical industrial CHP application for gas turbines is a chemicals plant with a 25 MW

[2] Fuel cells, which produce electricity from hydrogen and oxygen, emit only water vapor. There are emissions associated with producing the hydrogen supply depending on its source. However, most fuel cell technologies are still being developed, with only one type (phosphoric acid fuel cell) commercially available in limited production.
[3] PA Consulting Independent Power Database; Energy Nexus Group.

simple cycle gas turbine supplying base-load power to the plant with an unfired heat recovery steam generator (HRSG) on the exhaust. Approximately 29 MW thermal (MWth) of steam is produced for process use within the plant.

A typical commercial/institutional CHP application for gas turbines is a college or university campus with a 5 MW simple-cycle gas turbine. Approximately 8 MWth of 150 to 400 psig steam (or hot water) is produced in an unfired heat recovery steam generator and sent into a central thermal loop for campus space heating during winter months or to single-effect absorption chillers to provide cooling during the summer.

While the recovery of thermal energy provides compelling economics for gas turbine CHP, smaller gas turbines supply prime power in certain applications. Large industrial facilities install simple-cycle gas turbines without heat recovery to provide peaking power in capacity constrained areas, and utilities often place gas turbines in the 5 to 40 MW size range at substations to provide incremental capacity and grid support. A number of turbine manufacturers and packagers offer mobile turbine generator units in this size range that can be used in one location during a period of peak demand and then trucked to another location for the following season.

Technology Description

Basic Process and Components

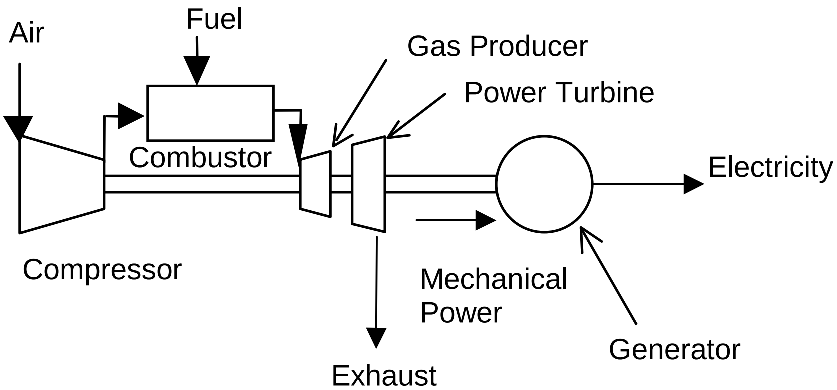
Gas turbine systems operate on the thermodynamic cycle known as the Brayton cycle. In a Brayton cycle, atmospheric air is compressed, heated, and then expanded, with the excess of power produced by the expander (also called the turbine) over that consumed by the compressor used for power generation. The power produced by an expansion turbine and consumed by a compressor is proportional to the absolute temperature of the gas passing through the device. Consequently, it is advantageous to operate the expansion turbine at the highest practical temperature consistent with economic materials and internal blade cooling technology and to operate the compressor with inlet air flow at as low a temperature as possible. As technology advances permit higher turbine inlet temperature, the optimum pressure ratio also increases.

Higher temperature and pressure ratios result in higher efficiency and specific power. Thus, the general trend in gas turbine advancement has been towards a combination of higher temperatures and pressures. While such advancements increase the manufacturing cost of the machine, the higher value, in terms of greater power output and higher efficiency, provides net economic benefits. The industrial gas turbine is a balance between performance and cost that results in the most economic machine for both the user and manufacturer.

Modes of Operation

There are several variations of the Brayton cycle in use today. Fuel consumption may be decreased by preheating the compressed air with heat from the turbine exhaust using a recuperator or regenerator; the compressor work may be reduced and net power increased by using intercooling or precooling; and the exhaust may be used to raise steam in a boiler and to generate additional power in a combined cycle. **Figure 2** shows the primary components of a simple cycle gas turbine.

Figure 2. Components of a Simple-Cycle Gas Turbine



Gas turbine exhaust is quite hot, up to 800 to 900°F for smaller industrial turbines and up to 1,100°F for some new, large central station utility machines and aeroderivative turbines. Such high exhaust temperatures permit direct use of the exhaust. With the addition of a heat recovery steam generator, the exhaust heat can produce steam or hot water. A portion or all of the steam generated by the HRSG may be used to generate additional electricity through a steam turbine in a combined cycle configuration.

A gas turbine based system is operating in combined heat and power mode when the waste heat generated by the turbine is applied in an end-use. For example, a simple-cycle gas turbine using the exhaust in a direct heating process is a CHP system, while a system that features all of the turbine exhaust feeding a HRSG and all of the steam output going to produce electricity in a combined-cycle steam turbine is not.

Types of Gas Turbines

Aeroderivative gas turbines for stationary power are adapted from their jet and turboshaft aircraft engine counterparts. While these turbines are lightweight and thermally efficient, they are usually more expensive than products designed and built exclusively for stationary applications. The largest aeroderivative generation turbines available are 40 to 50 MW in capacity. Many aeroderivative gas turbines for stationary use operate with compression ratios in the range of 30:1, requiring a high-pressure external fuel gas compressor. With advanced system developments, larger aeroderivative turbines (>40 MW) are approaching 45% simple-cycle efficiencies (LHV).

Industrial or frame gas turbines are exclusively for stationary power generation and are available in the 1 to 250 MW capacity range. They are generally less expensive, more rugged, can operate longer between overhauls, and are more suited for continuous base-load operation with longer inspection and maintenance intervals than aeroderivative turbines. However, they are less efficient and much heavier. Industrial gas turbines generally have more modest compression ratios (up to 16:1) and often do not require an external fuel gas compressor. Larger industrial gas turbines (>100 MW) are approaching simple-cycle efficiencies of approximately 40% (LHV) and combined-cycle efficiencies of 60% (LHV).

Industry uses gas turbines between 500 kW to 40 MW for on-site power generation and as mechanical drivers. Small gas turbines also drive compressors on long distance natural gas pipelines. In the petroleum industry turbines drive gas compressors to maintain well pressures and enable refineries and petrochemical plants to operate at elevated pressures. In the steel industry turbines drive air compressors used for blast furnaces. In process industries such as chemicals, refining and paper, and in large commercial and institutional applications turbines are used in combined heat and power mode generating both electricity and steam for use on-site.

Design Characteristics

Thermal output:	Gas turbines produce a high quality (high temperature) thermal output suitable for most combined heat and power applications. High-pressure steam can be generated or the exhaust can be used directly for process drying and heating.
Fuel flexibility:	Gas turbines operate on natural gas, synthetic gas, landfill gas, and fuel oils. Plants typically operate on gaseous fuel with a stored liquid fuel for backup to obtain the less expensive interruptible rate for natural gas.
Reliability and life:	Modern gas turbines have proven to be reliable power generators given proper maintenance. Time to overhaul is typically 25,000 to 50,000 hours.
Size range:	Gas turbines are available in sizes from 500 kW to 250 MW.
Emissions:	Many gas turbines burning gaseous fuels (mainly natural gas) feature lean premixed burners (also called dry low-NO_x combustors) that produce NO_x emissions below 25 ppm, with laboratory data down to 9 ppm, and simultaneous low CO emissions in the 10 to 50 ppm range.[4] Selective catalytic reduction (SCR) or catalytic combustion further reduces NO_x emissions. Many gas turbines sited in locales with stringent emission regulations use SCR after-treatment to achieve single-digit (below 9 ppm) NO_x emissions.

[4] Gas turbines have high oxygen content in their exhaust because they burn fuel with high excess air to limit combustion temperatures to levels that the turbine blades, combustion chamber and transition section can handle without compromising system life. Consequently, emissions from gas turbines are evaluated at a reference condition of 15% oxygen. For comparison, boilers use 3% oxygen as the reference condition for emissions, because they can minimize excess air and thus waste less heat in their stack exhaust. Note that due to the different amount of diluent gases in the combustion products, the mass of NO_x measured as 9 ppm @ 15% oxygen is approximately 27 ppm @ 3% oxygen, the condition used for boiler NO_x regulations.

Part-load operation:	Because gas turbines reduce power output by reducing combustion temperature, efficiency at part load can be substantially below that of full-power efficiency.

Performance Characteristics

Electrical Efficiency

The thermal efficiency of the Brayton cycle is a function of pressure ratio, ambient air temperature, turbine inlet air temperature, the efficiency of the compressor and turbine elements, turbine blade cooling requirements, and any performance enhancements (i.e., recuperation, intercooling, inlet air cooling, reheat, steam injection, or combined cycle). All of these parameters, along with gas turbine internal mechanical design features, have been improving with time. Therefore newer machines are usually more efficient than older ones of the same size and general type. The performance of a gas turbine is also appreciably influenced by the purpose for which it is intended. Emergency power units generally have lower efficiency and lower capital cost, while turbines intended for prime power, compressor stations and similar applications with high annual capacity factors having higher efficiency and higher capital costs. Emergency power units are permitted for a maximum number of hours per year and allowed to have considerably higher emissions than turbines permitted for continuous duty.

Table 1 summarizes performance characteristics for typical commercially available gas turbine CHP systems over the 1 to 40 MW size range. Heat rates shown are from manufacturers' specifications and industry publications. Available thermal energy (steam output) was calculated from published turbine data on turbine exhaust temperatures and flows. CHP steam estimates are based on an unfired HRSG with an outlet exhaust temperature of 280°F producing dry, saturated steam at 150 psig. Total efficiency is defined as the sum of the net electricity generated plus steam produced for plant thermal needs divided by total fuel input to the system. Higher steam pressures can be obtained but at slightly lower total efficiencies. Additional steam can be generated and total efficiency further increased with duct firing in the HRSG (see heat recovery section). To estimate fuel savings effective electrical efficiency is a more useful value than overall efficiency. Effective electric efficiency is calculated assuming the useful-thermal output from the CHP system would otherwsie be generated by an 80% efficient boiler. The theoretical boiler fuel is subtracted from the total fuel input and the remaining fuel input used to calculate the effective electric efficiency which can then be compared to traditional electric generation.

The data in the table show that electrical efficiency increases as combustion turbines become larger. As electrical efficiency increases, the absolute quantity of thermal energy available to produce steam decreases per unit of power output, and the ratio of power to heat for the CHP system increases. A changing ratio of power to heat impacts project economics and may affect the decisions that customers make in terms of CHP acceptance, sizing, and the desirability of selling power.

*Table 1. Gas Turbine CHP - Typical Performance Parameters**

Cost & Performance Characteristics[5]	System 1	System 2	System 3	System 4	System 5
Electricity Capacity (kW)	1,000	5,000	10,000	25,000	40,000
Total Installed Cost (2000 $/kW)[6]	$1,780	$1,010	$970	$860	$785
Electric Heat Rate (Btu/kWh), HHV[7]	15,580	12,590	11,765	9,945	9,220
Electrical Efficiency (%), HHV	21.9%	27.1%	29.0%	34.3%	37.0%
Fuel Input (MMBtu/hr)	15.6	62.9	117.7	248.6	368.8
Required Fuel Gas Pressure (psig)	95	160	250	340	435
CHP Characteristics					
Exhaust Flow (1,000 lb/hr)	44	162	316	571	954
GT Exhaust Temperature (Fahrenheit)	950	950	915	950	854
HRSG Exhaust Temperature (Fahrenheit)	280	280	280	280	280
Steam Output (MMBtu/hr)	7.1	26.6	49.6	95.6	136.8
Steam Output (1,000 lbs/hr)	6.7	25.0	46.6	89.8	128.5
Steam Output (kW equivalent)	2,080	7,800	14,540	28,020	40,100
Total CHP Efficiency (%), HHV[8]	68%	69%	71%	73%	74%
Power/Heat Ratio[9]	0.48	0.64	0.69	0.89	1.0
Net Heat Rate (Btu/kWh)[10]	6,673	5,947	5,562	5,164	4,944
Effective Electrical Efficiency (%), HHV[11]	51%	57%	61%	66%	69%

* *For typical systems commercially available in 2001*

Source: Energy Nexus Group[5]

Fuel Supply Pressure

Gas turbines need minimum gas pressure of about 100 psig for the smallest turbines with substantially higher pressures for larger turbines and aeroderivative machines. Depending on the supply pressure of the gas being delivered to the site the cost and power consumption of the fuel gas compressor can be a significant consideration. **Table 2** shows the power required to compress natural gas from supply pressures typical of commercial and industrial service to the pressures required by typical industrial gas turbines. Required supply pressures generally increase with gas turbine size.

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[5] Characteristics for "typical" commercially available gas turbine generator system. Data based on: Solar Turbines Saturn 20 – 1 MW; Solar Turbines Taurus 60 – 5 MW; Solar Turbines Mars 100 – 10 MW; GE LM2500+ – 25 MW; GE LM6000PD – 40 MW.

[6] Installed costs based on CHP system producing 150 psig saturated steam with an unfired heat recovery steam generator.

[7] All turbine and engine manufacturers quote heat rates in terms of the lower heating value (LHV) of the fuel. On the other hand, the usable energy content of fuels is typically measured on a higher heating value basis (HHV). In addition, electric utilities measure power plant heat rates in terms of HHV. For natural gas, the average heat content of natural gas is 1,030 Btu/scf on an HHV basis and 930 Btu/scf on an LHV basis – or about a 10% difference.

[8] Total Efficiency = (net electric generated + net steam produced for thermal needs)/total system fuel input

[9] Power/Steam Ratio = CHP electrical power output (Btu)/ useful steam output (Btu)

[10] Net Heat Rate = (total fuel input to the CHP system - the fuel that would be normally used to generate the same amount of thermal output as the CHP system output assuming an efficiency of 80%)/CHP electric output (kW).

[11] Effective Electrical Efficiency = (CHP electric power output)/(Total fuel into CHP system – total heat recovered/0.8); Equivalent to 3,412 Btu/kWh/Net Heat Rate.

Table 2. Power Requirements For Natural Gas Compression[12]

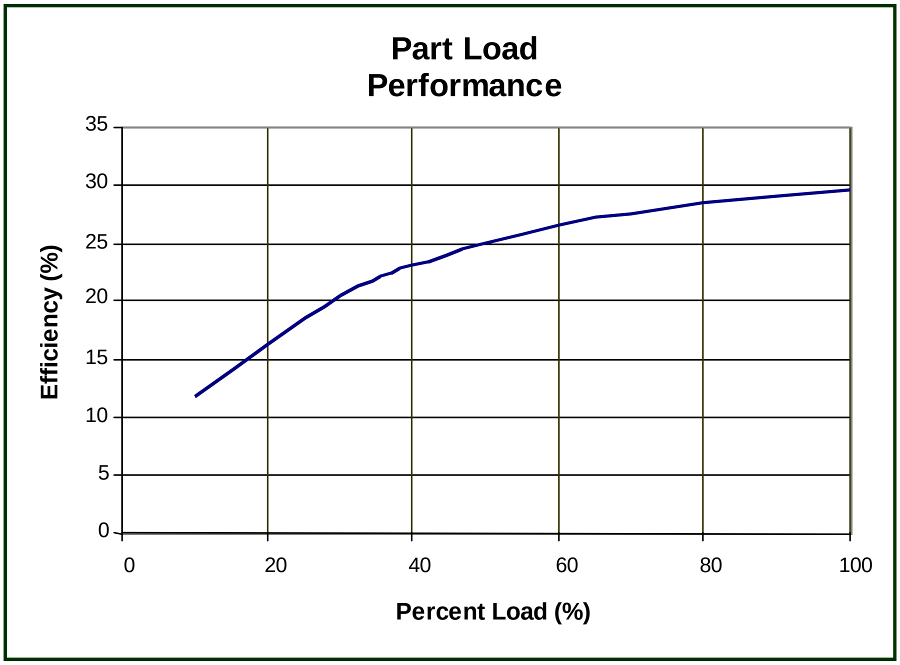
	System 1	System 2	System 3	System 4	System 5
Turbine Electric Capacity (kW)	1,000	5,000	10,000	25,000	40,000
Turbine Pressure Ratio	6.5	10.9	17.1	23.1	29.6
Required Compression Power (kW)					
50 psig gas supply pressure	17	125	310	650	1,310
150 psig gas supply pressure	NA	26	120	300	675
250 psig gas supply pressure	NA	NA	40	150	380

Source: Energy Nexus Group

Part-Load Performance

When less than full power is required from a gas turbine, the output is reduced by lowering the turbine inlet temperature. In addition to reducing power, this change in operating conditions also reduces efficiency. **Figure 3** shows a typical part-load derate curve. Emissions are generally increased at part load conditions, especially at half load and below.

Figure 3. Part Load Power Performance



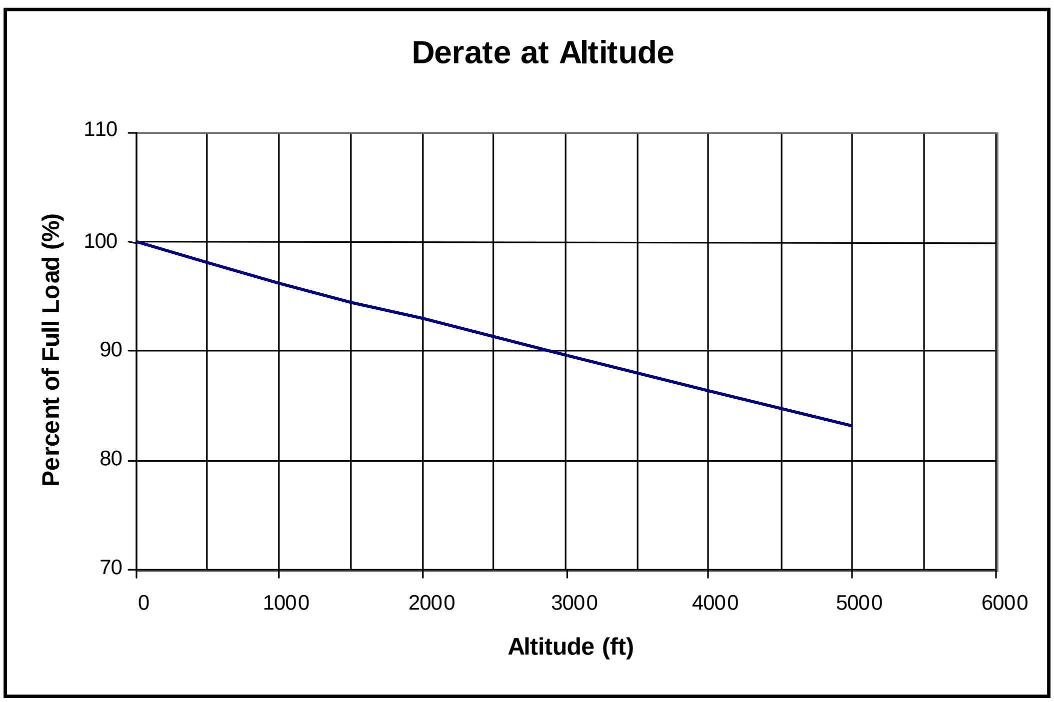
Source: Energy Nexus Group

[12] Fuel gas supply pressure requirements calculated assuming delivery of natural gas at an absolute pressure 35% greater than the compressor discharge in order to meet the requirements of the gas turbine flow control system and combustor mixing nozzles. Mass flow of fuel based on the fuel flow of reference gas turbines in the size range considered, and assuming an electric motor of 95% efficiency driving the booster compressor. Gas supply pressures of 50 psig, 150 psig and 250 psig form the basis of the calculations.

Effects of Ambient Conditions on Performance

The ambient conditions under which a gas turbine operates have a noticeable effect on both the power output and efficiency. At elevated inlet air temperatures, both the power and efficiency decrease. The power decreases due to the decreased air flow mass rate (the density of air declines as temperature increases) and the efficiency decreases because the compressor requires more power to compress air of higher temperature. Conversely, the power and efficiency increase when the inlet air temperature is reduced. **Figure 4** shows the variation in power and efficiency for a gas turbine as a function of ambient temperature compared to the reference International Organization for Standards (ISO) condition of sea level and 59°F. At inlet air temperatures of near 100°F, power output can drop to as low as 90% of ISO-rated power for typical gas turbines. At cooler temperatures of about 40 to 50°F, power can increase to as high as 105% of ISO-rated power.

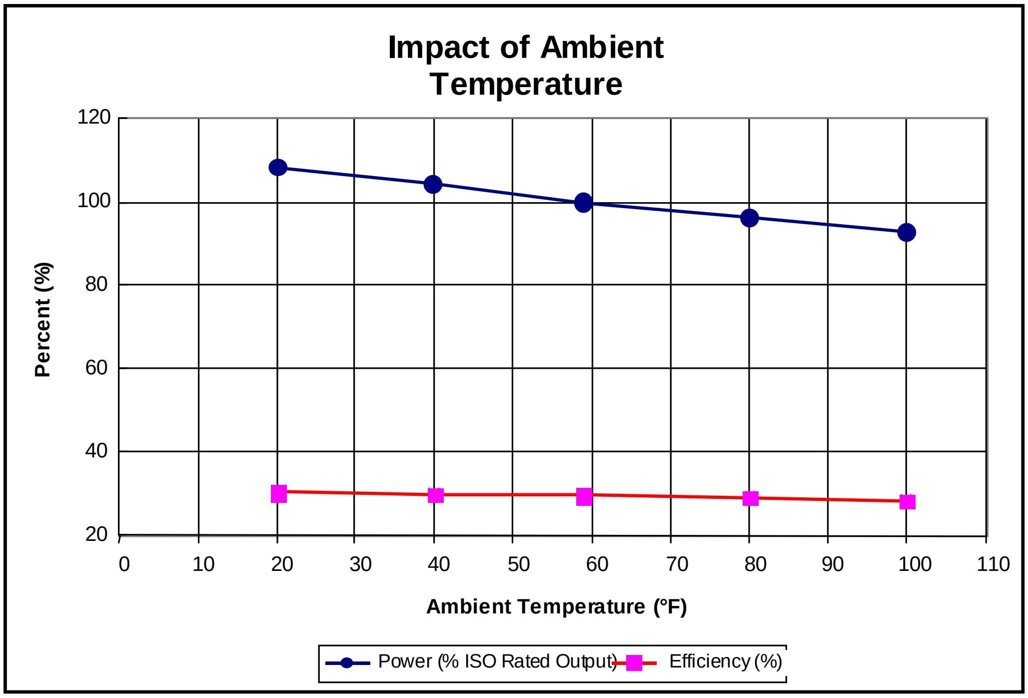
Figure 4. Ambient Temperature Effects on Performance



Source: Energy Nexus Group

The density of air decreases at altitudes above sea level. Consequently, power output decreases. **Figure 5** shows the impact of altitude derate.

Figure 5. Altitude Effects on Performance

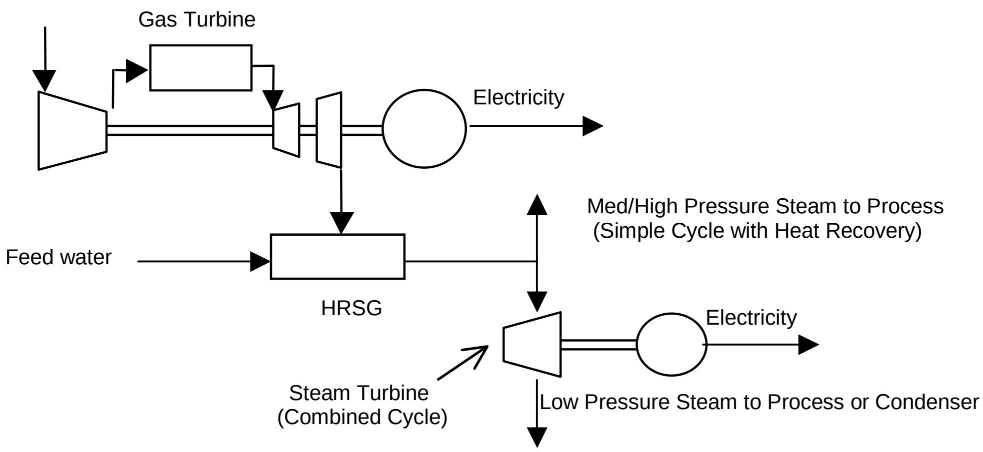


Source: Energy Nexus Group

Heat Recovery

The economics of gas turbines in process applications often depend on effective use of the thermal energy contained in the exhaust gas, which generally represents 60 to 70% of the inlet fuel energy. The most common use of this energy is for steam generation in unfired or supplementary fired heat recovery steam generators. However, the gas turbine exhaust gases can also be used as a source of direct process energy, for unfired or fired process fluid heaters, or as preheated combustion air for power boilers. **Figure 6** shows a typical gas turbine/HRSG configuration. An unfired HRSG is the simplest steam CHP configuration and can generate steam at conditions ranging from 150 psig to approximately 1,200 psig.

Figure 6. Heat Recovery from a Gas Turbine System



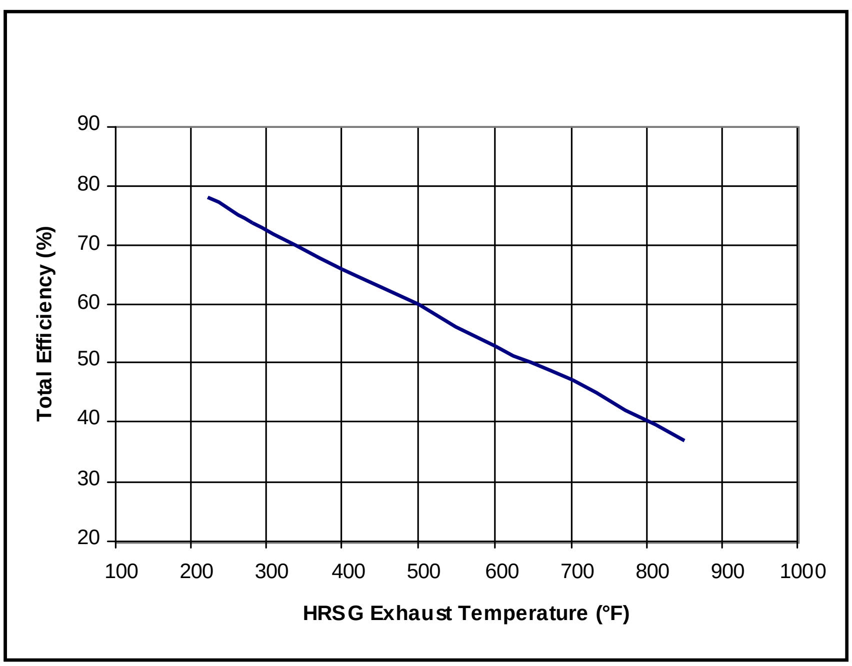
CHP System Efficiency

Overall or total efficiency of a CHP system is a function of the amount of energy recovered from the turbine exhaust. The two most important factors influencing the amount of energy available for steam generation are gas turbine exhaust temperature and HRSG stack temperature.

Turbine firing temperature and turbine pressure ratio combined determine gas turbine exhaust temperature. Typically aeroderivative gas turbines have higher firing temperatures than do industrial gas turbines, but when the higher pressure ratio of aeroderative gas turbines is recognized, the turbine discharge temperatures of the two turbine types remain somewhat close, typically in the range of 850 to 950°F. For the same HRSG exit temperature, higher turbine exhaust temperature (higher HRSG gas inlet temperature) results in greater available thermal energy and increased HRSG output.

Similarly, the lower the HRSG stack temperature, the greater the amount of energy recovered and the higher the total-system efficiency. HRSG stack temperature is a function of steam conditions and fuel type. Saturated steam temperatures increase with increasing steam pressure. Because of pinch point considerations within the HRSG, higher steam pressures result in higher HRSG exhaust stack temperatures, less utilization of available thermal energy, and a reduction in total CHP system efficiency. In general, minimum stack temperatures of about 300°F are recommended for sulfur bearing fuels. **Figure 7** illustrates the increase in overall system efficiency as the exhaust temperature decreases through effective heat recovery**.** Generally, unfired HRSGs can be designed to economically recover approximately 95% the available energy in the turbine exhaust (the energy released in going from turbine exhaust temperature to HRSG exhaust temperature).

Figure 7. Effect of Stack Temperature on Total CHP Efficiency*



** Based on an LM6000 with unfired HRSG*

Overall CHP efficiency generally remains high under part-load conditions. The decrease in electric efficiency from the gas turbine under part-load conditions results in a relative increase in heat available for recovery under these conditions. This can be a significant operating advantage for applications in which the economics are driven by high steam demand.

Supplementary Firing

Since the gas turbine combustion process consumes little of the available oxygen in the turbine air flow, the oxygen content in the gas turbine exhaust permits supplementary fuel firing ahead of the HRSG to increase steam production relative to an unfired unit. Supplementary firing can raise the exhaust gas temperature entering the HRSG up to 1,800°F and increase the amount of steam produced by the unit by a factor of two. Moreover, since the turbine exhaust gas is essentially preheated combustion air, the fuel consumed in supplementary firing is less than that required for a stand-alone boiler providing the same increment in steam generation. The HHV efficiency of incremental steam production from supplementary firing above that of an unfired HRSG is often 85% or more when firing natural gas.

Supplementary firing also increases system flexibility. Unfired HRSGs are typically convective heat exchangers that respond solely to exhaust conditions of the gas turbine and do not easily allow for steam flow control. Supplementary firing capability provides the ability to control steam production, within the capability of the burner system, independent of the normal gas turbine operating mode. Low NO_x duct burners with guaranteed emissions levels as low as 0.08 lb NO_x/MMBtu can be specified to minimize the NO_x contribution of supplemental firing.

Performance and Efficiency Enhancements

Recuperators

Several technologies that increase the output power and/or the efficiency of gas turbines have been developed and put into limited commercial service. Fuel use can be reduced (and hence efficiency improved) by use of a heat exchanger called a recuperator that uses the hot turbine exhaust to preheat the compressed air entering the combustor. Depending on gas turbine operating parameters, such a heat exchanger can add up to ten percentage points in machine efficiency (thereby raising efficiency from 30 to 40%). However, since there is increased pressure drop in both the compressed air and turbine exhaust sides of the recuperator, power output is typically reduced by 10 to 15%.

Recuperators are expensive, and their cost can normally only be justified when the gas turbine operates for a large number of full-power hours per year and the cost of fuel is relatively high. As an example, pipeline compressor station gas turbines frequently operate with high annual capacity factors, and some pipeline gas turbines have utilized recuperators since the 1960s. Recuperators also lower the temperature of the gas turbine exhaust, reducing the turbine’s effectiveness in CHP applications. Because recuperators are subject to large temperature differences, they are subject to significant thermal stresses. Cyclic operation in particular can fatigue joints, causing the recuperator to develop leaks and lose power and effectiveness. Design and manufacturing advancements have mitigated some of the cost and durability issues, and commercial recuperators have been introduced on microturbines and on a 4.2 MW industrial gas turbine (through a project supported by the U.S. Department of Energy).

Intercoolers

Intercoolers are used to increase gas turbine power by dividing the compressor into two sections and cooling the compressed air exiting the first section before it enters the second compressor section. Intercoolers reduce the power consumption in the second section of the compressor, thereby adding to the net power delivered by the combination of the turbine and compressor. Intercoolers have been used for decades on industrial air compressors and are used on some reciprocating engine turbochargers. Intercoolers generally are used where additional capacity is particularly valuable. Gas turbine efficiency does not change significantly with the use of intercooling. While intercoolers increase net output, the reduced power consumption of the second section of the compressor results in lower temperature for the compressed air entering the combustor and, consequently, incremental fuel is required.

Inlet Air Cooling

As shown in **Figure 4**, the decreased power and efficiency of gas turbines at high ambient temperatures means that gas turbine performance is at its lowest at the times power is often in greatest demand and most valued. The figure also shows that cooling the air entering the turbine by 40 to 50°F on a hot day can increase power output by 15 to 20%. The decreased power and efficiency resulting from high ambient air temperatures can be mitigated by any of several approaches to inlet-air cooling, including refrigeration, evaporative cooling, and thermal-energy storage using off-peak cooling.

With refrigeration cooling, either a compression driven or thermally activated (absorption chiller) refrigeration cycle cools the inlet air through a heat exchanger. The heat exchanger in the inlet air stream causes an additional pressure drop in the air entering the compressor, thereby slightly lowering cycle power and efficiency. However, as the inlet air is now substantially cooler than the ambient air there is a significant net gain in power and efficiency. Electric motor compression refrigeration requires a substantial parasitic power loss. Thermally activated absorption cooling can utilize waste heat from the gas turbine, reducing the direct parasitic loss. However, the complexity and cost of this approach pose potential drawbacks in many applications.

Evaporative cooling, which is widely used due to its low capital cost, uses a spray of water directly into the inlet air stream. Evaporation of the water reduces the temperature of the air. Since cooling is limited to the wet bulb air temperature, evaporative cooling is most effective when the wet bulb temperature is appreciably below the dry bulb (ordinary) temperature. Evaporative cooling can consume large quantities of water, making it difficult to operate in arid climates. A few large gas turbines have evaporative cooling, and it is expected to be used more frequently on smaller machines in the future.

The use of thermal-energy storage systems, typically ice, chilled water, or low-temperature fluids, to cool inlet air can eliminate most parasitic losses from the augmented power capacity. Thermal energy storage is a viable option if on-peak power pricing only occurs a few hours a day. In that case, the shorter time of energy storage discharge and longer time for daily charging allow for a smaller and less expensive thermal-energy storage system.

Capital Cost

A gas turbine CHP plant is a complex process with many interrelated subsystems. The basic package consists of the gas turbine, gearbox, electric generator, inlet and exhaust ducting, inlet air filtration, lubrication and cooling systems, standard starting system, and exhaust silencing. The basic package cost does not include extra systems such as the fuel-gas compressor, heat-recovery system, water-treatment system, or emissions-control systems such as selective catalytic reduction (SCR) or continuous emission monitoring systems (CEMS). Not all of these systems are required at every site. The cost of the basic turbine package plus the costs for added systems needed for the particular application comprise the total equipment cost. The total plant cost consists of total equipment cost plus installation labor and materials (including site work), engineering, project management (including licensing, insurance, commissioning, and startup), and financial carrying costs during the 6-18 month construction period.

Table 3 details estimated capital costs (equipment and installation costs) for the five typical gas turbine CHP systems**.** These are "typical" budgetary price levels; it should be noted that installed costs can vary significantly depending on the scope of the plant equipment, geographical area, competitive market conditions, special site requirements, emissions control requirements, prevailing labor rates, whether the system is a new or retrofit application, and whether or not the site is a green field or is located at an established industrial site with existing roads, water, fuel, electric, etc. The cost estimates presented in this section are based on systems that include DLE emissions control, unfired heat recovery steam generators (HRSG), fuel gas compression, water treatment for the boiler feed water, and basic utility interconnection for parallel power generation. There is no SCR system, no supplementary firing or duct burners, no building construction, and minimal site preparation and support.

The table shows that there are definite economies of scale for larger turbine power systems. Turbine packages themselves decline only slightly between the range of 5 to 40 MW, but ancillary equipment such as the HRSG, gas compression, water treatment, and electrical equipment are much lower in cost per unit of electrical output as the systems become larger.

Table 3. Estimated Capital Costs for Typical Gas Turbine-Based CHP Systems ($000s)[13]

Cost Component	System 1	System 2	System 3	System 4	System 5
Nominal Turbine Capacity (MW)	1	5	10	25	40
Equipment (Thousands of $)					
Turbine Genset	$675	$1,800	$4,000	$11,500	$15,800
Heat Recovery Steam Generators	$250	$450	$590	$1,020	$1,655
Water Treatment System	$30	$100	$150	$200	$225
Electrical Equipment	$150	$375	$625	$990	$1,500
Other Equipment	$145	$315	$575	$1,150	$1,875
Total Equipment	$1,250	$3,040	$5,940	$14,860	$21,055
Materials	$144	$346	$689	$1,490	$2,054
Labor	$348	$879	$1,752	$3,715	$4,723
Total Process Capital	$1,742	$4,265	$8,381	$20,065	$27,832
Project/Construction Management	$125	$304	$594	$1,486	$2,105
Engineering	$63	$153	$260	$537	$672
Project Contingency	$87	$215	$419	$1,005	$1,392
Project Financing	$129	$316	$618	$1,483	$2,048
Total Plant Cost	$2,146	$5,253	$10,272	$24,576	$34,049
Actual Turbine Capacity (kW)	1,210	5,200	10,600	28,600	43,400
Total Plant Cost per net kW ($)	$1,781	$1,010	$969	$859	$785

[13] Combustion turbine costs are based on published specifications and package prices. The total installed cost estimation is based in part on the use of a proprietary cost and performance model – SOAPP-CT.25 – (for state-of-the-art power plant, combustion turbine). The model output was adjusted based on Energy Nexus Group engineering judgment and experience and input from vendors and packagers. Actual costs can vary widely and are affected by site requirements and conditions, regional price variations, and environmental and other local permitting requirements.

Maintenance

Non-fuel operation and maintenance (O&M) costs presented in **Table 4** are based on gas turbine manufacturer estimates for service contracts, which consist of routine inspections and scheduled overhauls of the turbine generator set. Routine maintenance practices include on-line running maintenance, predictive maintenance, plotting trends, performance testing, fuel consumption, heat rate, vibration analysis, and preventive maintenance procedures. The O&M costs presented in **Table 4** include operating labor (distinguished between unmanned and 24 hour manned facilities) and total maintenance costs, including routine inspections and procedures and major overhauls.

Daily maintenance includes visual inspection by site personnel of filters and general site conditions. Routine inspections are required every 4,000 hours to insure that the turbine is free of excessive vibration due to worn bearings, rotors, and damaged blade tips. Inspections generally include on-site hot gas path boroscope inspections and non-destructive component testing using dye penetrant and magnetic particle techniques to ensure the integrity of components. The combustion path is inspected for fuel nozzle cleanliness and wear, along with the integrity of other hot gas path components.

A gas turbine overhaul is needed every 25,000 to 50,000 hours depending on service and is typically a complete inspection and rebuild of components to restore the gas turbine to nearly original or current (upgraded) performance standards. A typical overhaul consists of dimensional inspections, product upgrades and testing of the turbine and compressor, rotor removal, inspection of thrust and journal bearings, blade inspection and clearances and setting packing seals.

Gas turbine maintenance costs can vary significantly depending on the quality and diligence of the preventative maintenance program and operating conditions. Although gas turbines can be cycled, cycling every hour triples maintenance costs versus a turbine that operates for intervals of 1,000 hours or more. In addition, operating the turbine over the rated capacity for significant periods of time will dramatically increase the number of hot path inspections and overhauls. Gas turbines that operate for extended periods on liquid fuels will experience higher than average overhaul intervals.

Table 4. Gas Turbine Non-Fuel O&M Costs (Year 2000)

O&M Costs[14]	System 1	System 2	System 3	System 4	System 5
Electricity Capacity, kW	1,000	5,000	10,000	25,000	40,000
Variable (service contract), $/kWh	0.0045	0.0045	0.0045	0.0040	0.0035
Variable (consumables), $/kWh	0.0001	0.0001	0.0001	0.0001	0.0001
Fixed, $/kW-yr	40	10	7.5	6	5
Fixed, $/kWh @ 8,000 hrs/yr	0.0050	0.0013	0.0009	0.0008	0.0006
Total O&M Costs, $/kWh	0.0096	0.0059	0.0055	0.0049	0.0042

Fuels

All gas turbines intended for service as stationary power generators in the United States are available with combustors equipped to handle natural gas fuel. A typical range of heating values of gaseous fuels acceptable to gas turbines is 900 to 1,100 Btu per standard cubic foot (SCF), which covers the range of pipeline quality natural gas. Clean liquid fuels are also suitable for use in gas turbines.

Special combustors developed by some gas turbine manufacturers are capable of handling cleaned gasified solid and liquid fuels. Burners have been developed for medium Btu fuel (in the 400 to 500 Btu/SCF range), which is produced with oxygen-blown gasifiers, and for low Btu fuel (90 to 125 Btu/SCF), which is produced by air-blown gasifiers. These burners for gasified fuels exist for large gas turbines but are not available for small gas turbines.

Contaminants in fuel such as ash, alkalis (sodium and potassium), and sulfur result in alkali sulfate deposits, which impede flow, degrade performance, and cause corrosion in the turbine hot section. Fuels must have only low levels of specified contaminants in them (typically less than 10 ppm total alkalis, and single-digit ppm of sulfur).

Liquid fuels require their own pumps, flow control, nozzles and mixing systems. Many gas turbines are available with either gas or liquid firing capability. In general, gas turbines convert from one fuel to another quickly. Several gas turbines are equipped for dual firing and can switch fuels with minimal or no interruption.

Lean burn/dry low NO_x gas combustors generate NO_x emissions levels as low as 9 ppm (at 15% O_2). Liquid fuel combustors have NO_x emissions limited to approximately 25 ppm (at 15% O_2). There is no substantial difference in general performance with either fuel. However, the different heats of combustion result in slightly higher mass flows through the expansion turbine when liquid fuels are used, and thus result in a small increase in power and efficiency performance. In addition, the fuel pump work with liquid fuel is less than with the fuel gas booster compressor, thereby further increasing net performance with liquid fuels.

[14] O&M costs are based on 8,000 operating hours expressed in terms of annual electricity generation. Fixed costs are based on an interpolation of manufacturers' estimates. The variable component of the O&M cost represents the inspections and overhaul procedures that are normally conducted by the prime mover original equipment manufacturer through a service agreement usually based on run hours.

Gas turbines operate with combustors at pressure levels from 75 to 350 psig. While the pipeline pressure of natural gas is always above these levels, the pressure is normally let down during city gate metering and subsequent flow through the distribution piping system and customer metering. For example, local distribution gas pressures usually range from 30 to 130 psig in feeder lines and from 1 to 60 psig in final distribution lines. Depending on where the gas turbine is located on the gas distribution system, a fuel gas booster compressor may be required to ensure that fuel pressure is adequate for the gas turbine flow control and combustion systems. The cost of such booster compressors adds to the installation capital cost – fuel gas compressor costs can add from $20 to $150/kW to a CHP system's total cost, representing 2% of the total cost for a large system up to 10% of the total installed cost for a small gas turbine installation.[15] Redundant booster compressors ensure reliable operation because without adequate fuel pressure the gas turbine does not operate.

Availability

Many operational conditions affect the propensity to fail in a gas turbine. Frequent starts and stops incur damage from thermal cycling, which accelerates mechanical failure. Use of liquid fuels, especially heavy fuels and fuels with impurities (alkalis, sulfur, and ash), radiate heat to the combustor walls significantly more intensely than occurs with, clean, gaseous fuels, thereby overheating the combustor and transition piece walls. On the other hand, steady operation on clean fuels can permit gas turbines to operate for a year without need for shutdown. Estimated availability of gas turbines operating on clean gaseous fuels, like natural gas, is in excess of 95%.

Emissions

Gas turbines are among the cleanest fossil-fueled power generation equipment commercially available. Gas turbine emission control technologies are continuing to evolve, with older technologies gradually phasing out as new technologies are developed and commercialized.

The primary pollutants from gas turbines are oxides of nitrogen (NO_x), carbon monoxide (CO), and volatile organic compounds (VOCs). Other pollutants such as oxides of sulfur (SO_x) and particulate matter (PM) are primarily dependent on the fuel used. The sulfur content of the fuel determines emissions of sulfur compounds, primarily SO_2. Gas turbines operating on desulfized natural gas or distillate oil emit relatively insignificant levels of $SO_{x.}$ In general, SO_x emissions are greater when heavy oils are fired in the turbine. SO_x control is thus a fuel purchasing issue rather than a gas turbine technology issue. Particulate matter is a marginally significant pollutant for gas turbines using liquid fuels. Ash and metallic additives in the fuel may contribute to PM in the exhaust.

It is important to note that the gas turbine operating load has a significant effect on the emissions levels of the primary pollutants of NO_x, CO, and VOCs. Gas turbines typically operate at high loads. Consequently, gas turbines are designed to achieve maximum efficiency and optimum combustion conditions at high loads. Controlling all pollutants simultaneously at all load conditions is difficult. At higher loads, higher NO_x emissions occur due to peak flame

[15] American Gas Association, *Distributed Generation and the Natural Gas Infrastructure,* 1999

temperatures. At lower loads, lower thermal efficiencies and more incomplete combustion occurs resulting in higher emissions of CO and VOCs.

The pollutant referred to as NO_x is a mixture of mostly NO and NO_2 in variable composition. In emissions measurement it is reported as parts per million by volume in which both species count equally. NO_x is formed by three mechanisms: thermal NO_x, prompt NO_x, and fuel-bound NO_x. The predominant NO_x formation mechanism associated with gas turbines is thermal NO_x. Thermal NO_x is the fixation of atmospheric oxygen and nitrogen, which occurs at high combustion temperatures. Flame temperature and residence time are the primary variables that affect thermal NO_x levels. The rate of thermal NO_x formation increases rapidly with flame temperature. Prompt NO_x forms from early reactions of nitrogen modules in the combustion air and hydrocarbon radicals from the fuel. It forms within the flame and typically is about 1 ppm at 15% O_2, and is usually much smaller than the thermal NO_x formation. Fuel-bound NO_x forms when the fuel contains nitrogen as part of the hydrocarbon structure. Natural gas has negligible chemically bound fuel nitrogen.

The control of peak flame temperature, through diluent (water or steam) injection or by maintaining homogenous fuel-to-air ratios that keep local flame temperature below the stoichiometric adiabatic temperature, have been the traditional methods of limiting NO_x formation. In older diffusion flame combustion systems, fuel/air mixing and combustion occurred simultaneously. This resulted in local fuel/air mixture chemical concentrations that produced high local flame temperatures. These high temperature “hot spots” are where most of the NO_x emissions originate. Many new gas turbines feature lean pre-mixed combustion systems. These systems, sometimes referred to as dry low NO_x (DLN) or dry low emissions (DLE), operate in a tightly controlled lean (lower fuel-to-air ratio) premixed mode that maintains modest peak flame temperatures.

CO and VOCs both result from incomplete combustion. CO emissions result when there is insufficient residence time at high temperature. In gas turbines, the failure to achieve CO burnout may result from the quenching effects of dilution and combustor wall cooling air. CO emissions are also heavily dependent on the operating load of the turbine. For example, a gas turbine operating under low loads will tend to have incomplete combustion, which will increase the formation of CO. CO is usually regulated to levels below 50 ppm for both health and safety reasons. Achieving such low levels of CO had not been a problem until manufacturers achieved low levels of NOx, because the techniques used to engineer DLN combustors had a secondary effect of increasing CO emissions.

VOCs can encompass a wide range of compounds, some of which are hazardous air pollutants. These compounds discharge into the atmosphere when some portion of the fuel remains unburned or just partially burned. Some organics are unreacted trace constituents of the fuel, while others may be pyrolysis products of the heavier hydrocarbons in the gas.

While not considered a regulated pollutant in the ordinary sense of directly affecting public health, emissions of carbon dioxide (CO_2) are of concern due to its contribution to global warming. Atmospheric warming occurs because solar radiation readily penetrates to the surface of the planet but infrared (thermal) radiation from the surface is absorbed by the CO_2 (and other polyatomic gases such as methane, unburned hydrocarbons, refrigerants, water vapor, and volatile chemicals) in the atmosphere, with resultant increase in temperature of the atmosphere. The amount of CO_2 emitted is a function of both fuel carbon content and system efficiency. The fuel carbon content of natural gas is 34 lbs carbon/MMBtu; oil is 48 lbs carbon/MMBtu; and (ash-free) coal is 66 lbs carbon/MMBtu.

Emissions Control Options

NO_x control has been the primary focus of emission control research and development in recent years. The following provides a description of the most prominent emission control approaches:

Diluent Injection

The first technique used to reduce NO_x emissions was injection of water or steam into the high temperature zones of the flame. Water and steam are strong diluents and can quench hot spots in the flame reducing NO_x. However, positioning of the injection is not precise and some NO_x is still created. Depending on uncontrolled NO_x levels, water or steam injection reduces NO_x by 60% or more. Water or steam injection enables gas turbines to operate with NO_x levels as low as 25 ppm (@ 15% O_2) on natural gas. NO_x is only reduced to 42 to 75 ppm when firing with liquid distillate fuel. Both water and steam increase the mass flow through the turbine and create a small amount of additional power. Use of exhaust heat to raise the steam temperature also increases overall efficiency slightly. The water used needs to be demineralized thoroughly in order to avoid forming deposits and corrosion in the turbine expansion section. This adds cost and complexity to the operation of the turbine. Diluent injection increases CO emissions appreciably as it lowers the temperature in the burnout zone as well as well as in the NO_x formation zone.

Lean Premixed Combustion

As discussed earlier, thermal NO_x formation is a function of both flame temperature and residence time. The focus of combustion improvements of the past decade was to lower flame hot spot temperature using lean fuel/air mixtures. Lean combustion decreases the fuel/air ratio in the zones where NO_x production occurs so that peak flame temperature is less than the stoichiometric adiabatic flame temperature, therefore suppressing thermal NO_x formation.

Lean premixed combustion (DLN/DLE) pre-mixes the gaseous fuel and compressed air so that there are no local zones of high temperatures, or "hot spots," where high levels of NO_x would form. Lean premixed combustion requires specially designed mixing chambers and mixture inlet zones to avoid flashback of the flame. Optimized application of DLN combustion requires an integrated approach to combustor and turbine design. The DLN combustor becomes an intrinsic part of the turbine design, and specific combustor designs must be developed for each turbine application. While NO_x levels as low as 9 ppm have been achieved with lean premixed combustion, few DLN equipped turbines have reached the level of practical operation at this emissions level necessary for commercialization – the capability of maintaining 9 ppm across a wide operating range from full power to minimum load. One problem is that pilot flames, which are small diffusion flames and a source of NO_x, are usually used for continuous internal ignition and stability in DLN combustors and make it difficult to maintain full net NO_x reduction over the complete turndown range.

Noise can also be an issue in lean premixed combustors as acoustic waves form due to combustion instabilities when the premixed fuel and air ignite. This noise also manifests itself as pressure waves, which can damage combustor walls and accelerate the need for combustor replacement, thereby adding to maintenance costs and lowering unit availability.

All leading gas turbine manufacturers feature DLN combustors in at least parts of their product lines. Turbine manufacturers generally guarantee NO_x emissions of 15 to 42 ppm using this technology. NO_x emissions when firing distillate oil are typically guaranteed at 42 ppm with DLN and/or combined with water injection. A few models (primarily those larger than 40 MW) have combustors capable of 9 ppm (natural gas fired) over the range of expected operation.

The development of market-ready DLN equipped turbine models is an expensive undertaking because of the operational difficulties in maintaining reliable gas turbine operation over a broad power range. Therefore, the timing of applying DLN to multiple turbine product lines is a function of market priorities and resource constraints. Gas turbine manufacturers initially develop DLN combustors for the gas turbine models for which they expect the greatest market opportunity. As time goes on and experience is gained, the technology is extended to additional gas turbine models.

Selective Catalytic Reduction

The primary post-combustion NO_x control method in use today is selective catalytic reduction (SCR). Ammonia is injected into the flue gas and reacts with NO_x in the presence of a catalyst to produce N_2 and H_2O. The SCR system is located in the exhaust path, typically within the HRSG where the temperature of the exhaust gas matches the operating temperature of the catalyst. The operating temperature of conventional SCR systems ranges from 400 to 800°F. The cost of conventional SCR has dropped significantly over time -- catalyst innovations have been a principal driver, resulting in a 20% reduction in catalyst volume and cost with no change in performance.

Low temperature SCR, operating in the 300 to 400°F temperature range, was commercialized in 1995 and is currently in operation on approximately twenty gas turbines. Low temperature SCR is ideal for retrofit applications where it can be located downstream of the HRSG, avoiding the potentially expensive retrofit of the HRSG to locate the catalyst within a hotter zone of the HRSG.

High temperature SCR installations, operating in the 800 to 1,100°F temperature range, have increased significantly in recent years. The high operating temperature permits the placement of the catalyst directly downstream of the turbine exhaust flange. High temperature SCR is also used on peaking capacity and base-loaded simple-cycle gas turbines where there is no HRSG.

SCR reduces between 80 to 90% of the NO_x in the gas turbine exhaust, depending on the degree to which the chemical conditions in the exhaust are uniform. When used in series with water/steam injection or DLN combustion, SCR can result in low single digit NO_x levels (2 to 5 ppm).

SCR systems are expensive and significantly impact the economic feasibility of smaller gas turbine projects. For a 5 MW project electric generation costs increase approximately half a cent per kWh.[16] SCR requires on-site storage of ammonia, a hazardous chemical. In addition, ammonia can "slip" through the process unreacted, contributing to environmental health concerns.[17]

Carbon Monoxide Oxidation Catalysts

Oxidation catalysts control CO in gas turbine exhaust. Some SCR installations incorporate CO oxidation modules along with NO_x reduction catalysts for simultaneous control of CO and NO_x. The CO catalyst promotes the oxidation of CO and hydrocarbon compounds to carbon dioxide (CO_2) and water (H_20) as the exhaust stream passes the through the catalyst bed. The oxidation process takes place spontaneously so no reactants are required. The catalyst is usually made of precious metal such as platinum, palladium, or rhodium. Other formations, such as metal oxides for emission streams containing chlorinated compounds, are also used. CO catalysts also reduce VOCs and organic hazardous air pollutants (HAPs). CO catalysts on gas turbines result in approximately 90% reduction of CO and 85 to 90% control of formaldehyde (similar reductions can be expected on other HAPs).

Catalytic Combustion

In catalytic combustion, fuels oxidize at lean conditions in the presence of a catalyst. Catalytic combustion is a flameless process, allowing fuel oxidation to occur at temperatures below 1,700°F, where NO_x formation is low. The catalyst is applied to combustor surfaces, which cause the fuel air mixture to react with the oxygen and release its initial thermal energy. The combustion reaction in the lean premixed gas then goes to completion at design temperature. Data from ongoing long term testing indicates that catalytic combustion exhibits low vibration and acoustic noise, only one-tenth to one-hundredth the levels measured in the same turbine equipped with DLN combustors.

Gas turbine catalytic combustion technology is being pursued by developers of combustion systems and gas turbines and by government agencies, most notably the U.S. Department of Energy and the California Energy Commission. Past efforts at developing catalytic combustors for gas turbines achieved low, single-digit NO_x ppm levels, but failed to produce combustion systems with suitable operating durability. This was typically due to cycling damage and to the brittle nature of the materials used for catalysts and catalyst support systems. Catalytic combustor developers and gas turbine manufacturers are testing durable catalytic and "partial catalytic" systems that are overcoming the problems of past designs. Catalytic combustors capable of achieving NO_x levels below 3 ppm are in full-scale demonstration and are entering

[16] *Cost Analysis of NOx Control Alternatives for Stationary Gas Turbines*, ONSITE SYCOM Energy Corporation, November, 1999.

[17] The SCR reaction, with stoichiometric (for NO_x reduction) ammonia or other reagent should eliminate all NO_x. However because of imperfect mixing in the combustor the NO_x is not uniformly distributed across the turbine exhaust. Additionally the ammonia, or other reagent, also is not injected in a precisely uniform manner. These two non-uniformities in chemical composition cause either excess ammonia to be used, and to consequently "slip" out the exhaust, or for incomplete reaction of the NO_x in the turbine exhaust.

early commercial introduction.[18] Similarly to DLN combustion, optimized catalytic combustion requires an integrated approach to combustor and turbine design. Catalytic combustors must be tailored to the specific operating characteristics and physical layout of each turbine design.

Catalytic Absorption Systems

SCONOx™, patented by Goaline Environmental Technologies (currently EmerChem), is a post-combustion alternative to SCR that reduces NOx emissions to less than 2.5 ppm and almost 100% removal of CO. SCONOx™ combines catalytic conversion of CO and NOx with an absorption/regeneration process that eliminates the ammonia reagent found in SCR technology. It is based on a unique integration of catalytic oxidation and absorption technology. CO and NO catalytically oxidize to CO_2 and NO_2. The NO_2 molecules are subsequently absorbed on the treated surface of the SCONOx™ catalyst. The system does not require the use of ammonia, eliminating the potential for ammonia slip associated with SCR. The SCONOx™ system is generally located within the HRSG and under special circumstances may be located downstream of the HRSG. The system operates between 300-700ºF. U.S. EPA Region 9 identified SCONOx™ as "Lowest Achievable Emission Rate (LAER)" technology for gas turbine NO_x control in 1998.

The SCONOx™ technology is still in the early stages of market introduction. Issues that may impact application of the technology include relatively high capital cost, large reactor size compared to SCR, system complexity, high utilities cost and demand (steam, natural gas, compressed air and electricity are required), and a gradual rise in NO emissions over time that requires a 1 to 2 day shutdown every 6 to 12 months (depending on fuel quality and operation) to remove and regenerate the absorption modules ex-situ.[19]

Gas Turbine Emissions Characteristics

Table 5 shows typical emissions for each of the five typical turbine systems for the base year (2000). Typical emissions presented are based on gas turbine exhaust with no exhaust treatment and reflect what manufacturers will guarantee. Notable outliers for specific installations or engine models are identified. Due to the uniqueness of the combustion system of each gas turbine model, clear distinctions need to be made when discussing emissions technology and the corresponding emissions levels. Those distinctions are technology that is commercially available, technology that is technically proven but not yet commercial, and technology that is technically feasible but neither technically proven nor commercially available. This is particularly true for pollution prevention and combustion technologies as opposed to exhaust treatment control alternatives. The later two distinctions do not fall under the category of commercially available and consequently are noted as footnotes in **Table 5** rather than the representative emissions level.

Add-on control options for NO_x and CO can further reduce emissions of each by 80 to 90%. For many distributed generation gas turbine installations, exhaust treatment options have for the most part been avoided or not implemented due to the unfavorable capital and operating costs impacts.

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[18] For example, Kawasaki offers a version of their M1A 13X, 1.4 MW gas turbine with a catalytic combustor with less than 3 ppm NOx guaranteed.
[19] Resource Catalysts, Inc.

Table 5. Gas Turbine Emissions Characteristics Without Heat Recovery or Exhaust Control Options*

Emissions Characteristics	System 1	System 2	System 3	System 4	System 5
Electricity Capacity (kW)	1,000	5,000	10,000	25,000	40,000
Electrical Efficiency (HHV)	22%	27%	29%	34%	37%
NO_x, ppm	42[20]	25[21,]	25	25	25[22]
NO_x, lb/MWh[23]	2.43	1.16	1.08	0.92	0.31
CO, ppmv[24]	20	20	20	20	20
CO, lb/MWh[17]	0.71	0.56	0.53	0.45	0.85
CO_2, lb/MWh	1,887	1,510	1,411	1,193	1,106
Carbon, lb/MWh	515	412	385	326	302

* *For typical systems commercially available in 2001. Emissions estimates for untreated turbine exhaust conditions (15% O_2, no SCR or other exhaust clean up). Estimates based on typical manufacturers' guarantees using commercially available dry low NOx combustion technology.*

[20] 42 ppm represents the representative guaranteed state-of -the-art for 1 MW gas turbine systems. It has just been announced that Kawasaki is offering their M1A-13A equipped with a Xonon catalytic combustion system provided by Catalytica Energy Systems with a guarantee of less than 3 ppm NO_x. Many of the models in this size range do not have DLN options and still utilize diffusion flame combustion systems.

[21] Solar Turbines has permitted a 5 MW turbine guaranteed for 15 ppm NO_x. This specific gas turbine installation is equipped with a developmental ceramic combustor liner that is not standard on the commercial product line.

[22] 9 ppm is offered on industrial frame machines (e.g., GE 6B, Alstom GT10) which have lower firing temperatures and pressure ratios (resulting in lower efficiencies), and longer residence time.

[23] Conversion from volumetric emission rate (ppm at 15% O_2) to output based rate (lbs/MWh) for both NO_x and CO based on conversion multipliers provided by Catalytica Energy Systems (*http://www.catalyticaenergy.com/xonon/emissions_factors.html*).

[24] CO catalytic oxidation modules on gas turbines result in approximately 90% reduction of CO. Recent permits have included the utilization of CO catalysts to achieve less than 5 ppm CO.

Technology Characterization: Microturbines

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Technology Characterization – Microturbines

Introduction and Summary

Microturbines are small electricity generators that burn gaseous and liquid fuels to create high-speed rotation that turns an electrical generator. Today's microturbine technology is the result of development work in small stationary and automotive gas turbines, auxiliary power equipment, and turbochargers, much of which was pursued by the automotive industry beginning in the 1950s. Microturbines entered field testing around 1997 and began initial commercial service in 2000.

The size range for microturbines available and in development is from 30 to 350 kilowatts (kW), while conventional gas turbine sizes range from 500 kW to 250 megawatts (MW). Microturbines run at high speeds and, like larger gas turbines, can be used in power-only generation or in combined heat and power (CHP) systems. They are able to operate on a variety of fuels, including natural gas, sour gases (high sulfur, low Btu content), and liquid fuels such as gasoline, kerosene, and diesel fuel/distillate heating oil. In resource recovery applications, they burn waste gases that would otherwise be flared or released directly into the atmosphere.

Applications

Microturbines are ideally suited for distributed generation applications due to their flexibility in connection methods, ability to be stacked in parallel to serve larger loads, ability to provide stable and reliable power, and low emissions. Types of applications include:

- Peak shaving and base load power (grid parallel)
- Combined heat and power
- Stand-alone power
- Backup/standby power
- Ride-through connection
- Primary power with grid as backup
- Microgrid
- Resource recovery

Target customers include financial services, data processing, telecommunications, restaurant, lodging, retail, office building, and other commercial sectors. Microturbines are currently operating in resource recovery operations at oil and gas production fields, wellheads, coal mines, and landfill operations, where byproduct gases serve as essentially free fuel. Reliable unattended operation is important since these locations may be remote from the grid, and even when served by the grid, may experience costly downtime when electric service is lost due to weather, fire, or animals.

In CHP applications, the waste heat from the microturbine is used to produce hot water, to heat building space, to drive absorption cooling or desiccant dehumidification equipment, and to supply other thermal energy needs in a building or industrial process.

Technology Description

Basic Processes

Microturbines are small gas turbines, most of which feature an internal heat exchanger called a recuperator. In a microturbine, a radial flow (centrifugal) compressor compresses the inlet air that is then preheated in the recuperator using heat from the turbine exhaust. Next, the heated air from the recuperator mixes with fuel in the combustor and hot combustion gas expands through the expansion and power turbines. The expansion turbine turns the compressor and, in single-shaft models, turns the generator as well. Two-shaft models use the compressor drive turbine's exhaust to power a second turbine that drives the generator. Finally, the recuperator uses the exhaust of the power turbine to preheat the air from the compressor.

Single-shaft models generally operate at speeds over 60,000 revolutions per minute (rpm) and generate electrical power of high frequency, and of variable frequency (alternating current --AC). This power is rectified to direct current (DC) and then inverted to 60 hertz (Hz) for U.S. commercial use. In the two-shaft version, the power turbine connects via a gearbox to a generator that produces power at 60 Hz. Some manufacturers offer units producing 50 Hz for use in countries where 50 Hz is standard, such as in Europe and parts of Asia.

Thermodynamic Cycle

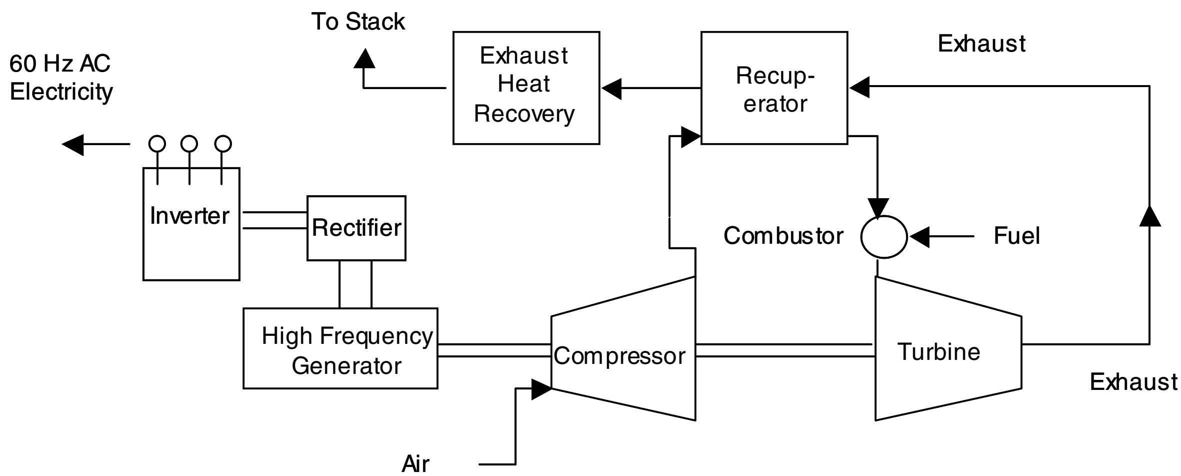
Microturbines operate on the same thermodynamic cycle, known as the Brayton cycle, as larger gas turbines. In this cycle, atmospheric air is compressed, heated, and then expanded, with the excess power produced by the expander (also called the turbine) over that consumed by the compressor used for power generation. The power produced by an expansion turbine and consumed by a compressor is proportional to the absolute temperature of the gas passing through those devices. Consequently, it is advantageous to operate the expansion turbine at the highest practical temperature consistent with economic materials and to operate the compressor with inlet airflow at as low a temperature as possible. As technology advances permit higher turbine inlet temperature, the optimum pressure ratio also increases. Higher temperature and pressure ratios result in higher efficiency and specific power. Thus, the general trend in gas turbine advancement has been towards a combination of higher temperatures and pressures. However, microturbine inlet temperatures are generally limited to 1,800ºF or below to enable the use of relatively inexpensive materials for the turbine wheel, and to maintain pressure ratios at a comparatively low 3.5 to 4.0.

Basic Components

Turbo-Compressor Package

The basic components of a microturbine are the compressor, turbine generator, and recuperator (*see* **Figure 1**). The heart of the microturbine is the compressor-turbine package, which is commonly mounted on a single shaft along with the electric generator. Two bearings support the single shaft. The single moving part of the one-shaft design has the potential for reducing maintenance needs and enhancing overall reliability. There are also two-shaft versions, in which the turbine on the first shaft directly drives the compressor while a power turbine on the second shaft drives a gearbox and conventional electrical generator producing 60 Hz power. The two-shaft design features more moving parts but does not require complicated power electronics to convert high frequency AC power output to 60 Hz.

Figure 1. Microturbine-Based CHP System (Single-Shaft Design)



Moderate to large-size gas turbines use multi-stage axial flow turbines and compressors, in which the gas flows along the axis of the shaft and is compressed and expanded in multiple stages. However, microturbine turbomachinery is based on single-stage radial flow compressors and turbines. Radial flow turbomachinery handles the small volumetric flows of air and combustion products with reasonably high component efficiency.[1] Large-size axial flow turbines and compressors are typically more efficient than radial flow components. However, in the size range of microturbines -- 0.5 to 5 lbs/second of air/gas flow -- radial flow components offer minimum surface and end wall losses and provide the highest efficiency.

In microturbines, the turbocompressor shaft generally turns at high rotational speed, about 96,000 rpm in the case of a 30 kW machine and about 80,000 rpm in a 75 kW machine. One 45 kW model on the market turns at 116,000 rpm. There is no single rotational speed-power size rule, as the specific turbine and compressor design characteristics strongly influence the physical size of components and consequently rotational speed. For a specific aerodynamic design, as the power rating decreases, the shaft speed increases, hence the high shaft speed of the small microturbines.

The radial flow turbine-driven compressor is quite similar in terms of design and volumetric flow to automobile, truck, and other small reciprocating engine turbochargers. Superchargers and turbochargers have been used for almost 80 years to increase the power of reciprocating engines

[1] With axial flow turbomachinery, blade height would be too small to be practical.

by compressing the inlet air to the engine. Today's world market for small automobile and truck turbochargers is around two million units per year. Small gas turbines, of the size and power rating of microturbines, serve as auxiliary power systems on airplanes. Cabin cooling (air conditioning) systems of airplanes use this same size and design family of compressors and turbines. The decades of experience with these applications provide the basis for the engineering and manufacturing technology of microturbine components.

Generator

The microturbine produces electrical power either via a high-speed generator turning on the single turbo-compressor shaft or with a separate power turbine driving a gearbox and conventional 3,600 rpm generator. The high-speed generator of the single-shaft design employs a permanent magnet (typically Samarium-Cobalt) alternator, and requires that the high frequency AC output (about 1,600 Hz for a 30 kW machine) be converted to 60 Hz for general use. This power conditioning involves rectifying the high frequency AC to DC, and then inverting the DC to 60 Hz AC. Power conversion comes with an efficiency penalty (approximately five percent). To start-up a single shaft design, the generator acts as a motor turning the turbo-compressor shaft until sufficient rpm is reached to start the combustor. Full start-up requires several minutes. If the system is operating independent of the grid (black starting), a power storage unit (typically a battery UPS) is used to power the generator for start-up.

Recuperators

Recuperators are heat exchangers that use the hot turbine exhaust gas (typically around 1,200ºF) to preheat the compressed air (typically around 300ºF) going into the combustor, thereby reducing the fuel needed to heat the compressed air to turbine inlet temperature. Depending on microturbine operating parameters, recuperators can more than double machine efficiency. However, since there is increased pressure drop in both the compressed air and turbine exhaust sides of the recuperator, power output typically declines 10 to 15% from that attainable without the recuperator. Recuperators also lower the temperature of the microturbine exhaust, reducing the microturbine's effectiveness in CHP applications.

Bearings

Microturbines operate on either oil-lubricated or air bearings, which support the shaft(s). *Oil-lubricated bearings* are mechanical bearings and come in three main forms – high-speed metal roller, floating sleeve, and ceramic surface. The latter typically offer the most attractive benefits in terms of life, operating temperature, and lubricant flow. While they are a well-established technology, they require an oil pump, oil filtering system, and liquid cooling that add to microturbine cost and maintenance. In addition, the exhaust from machines featuring oil-lubricated bearings may not be useable for direct space heating in cogeneration configurations due to the potential for contamination. Since the oil never comes in direct contact with hot combustion products, as is the case in small reciprocating engines, it is believed that the reliability of such a lubrication system is more typical of ship propulsion diesel systems (which have separate bearings and cylinder lubrication systems) and automotive transmissions than cylinder lubrication in automotive engines.

Air bearings have been in service on airplane cabin cooling systems for many years. They allow the turbine to spin on a thin layer of air, so friction is low and rpm is high. No oil or oil pump is needed. Air bearings offer simplicity of operation without the cost, reliability concerns, maintenance requirements, or power drain of an oil supply and filtering system. Concern does exist for the reliability of air bearings under numerous and repeated starts due to metal on metal friction during startup, shutdown, and load changes. Reliability depends largely on individual manufacturers' quality control methodology more than on design engineering, and will only be proven after significant experience with substantial numbers of units with long numbers of operating hours and on/off cycles.

Power Electronics

As discussed, single-shaft microturbines feature digital power controllers to convert the high frequency AC power produced by the generator into usable electricity. The high frequency AC is rectified to DC, inverted back to 60 or 50 Hz AC, and then filtered to reduce harmonic distortion. This is a critical component in the single-shaft microturbine design and represents significant design challenges, specifically in matching turbine output to the required load. To allow for transients and voltage spikes, power electronics designs are generally able to handle seven times the nominal voltage. Most microturbine power electronics are generating three-phase electricity.

Electronic components also direct all of the operating and startup functions. Microturbines are generally equipped with controls that allow the unit to be operated in parallel or independent of the grid, and internally incorporate many of the grid and system protection features required for interconnect. The controls also allow for remote monitoring and operation.

CHP Operation

In CHP operation, a second heat exchanger, the exhaust gas heat exchanger, transfers the remaining energy from the microturbine exhaust to a hot water system. Exhaust heat can be used for a number of different applications, including potable water heating, driving absorption cooling and desiccant dehumidification equipment, space heating, process heating, and other building or site uses. Some microturbine-based CHP applications do not use recuperators. With these microturbines, the temperature of the exhaust is higher and thus more heat is available for recovery. **Figure 1** illustrates a microturbine-based CHP system.

Design Characteristics

Thermal output:	Microturbines produce thermal output at temperatures in the 400 to 600°F range, suitable for supplying a variety of building thermal needs.
Fuel flexibility:	Microturbines can operate using a number of different fuels: natural gas, sour gases (high sulfur, low Btu content), and liquid fuels such as gasoline, kerosene, and diesel fuel/heating oil.

Reliability and life:	Design life is estimated to be in the 40,000 to 80,000 hour range. While units have demonstrated reliability, they have not been in commercial service long enough to provide definitive life data.
Size range:	Microturbines available and under development are sized from 30 to 350 kW.
Emissions:	Low inlet temperatures and high fuel-to-air ratios result in NO_x emissions of less than 10 parts per million (ppm) when running on natural gas.
Modularity:	Units may be connected in parallel to serve larger loads and provide power reliability.
Part-load operation:	Because microturbines reduce power output by reducing mass flow and combustion temperature, efficiency at part load can be below that of full-power efficiency.
Dimensions:	About 12 cubic feet.

Performance Characteristics

Microturbines are more complex than conventional simple-cycle gas turbines, as the addition of the recuperator both reduces fuel consumption (thereby substantially increasing efficiency) and introduces additional internal pressure losses that moderately lower efficiency and power. As the recuperator has four connections -- to the compressor discharge, the expansion turbine discharge, the combustor inlet, and the system exhaust -- it becomes a challenge to the microturbine product designer to make all of the connections in a manner that minimizes pressure loss, keeps manufacturing cost low, and entails the least compromise of system reliability. Each manufacturer's models have evolved in unique ways.

The addition of a recuperator opens numerous design parameters to performance-cost tradeoffs. In addition to selecting the pressure ratio for high efficiency and best business opportunity (high power for low price), the recuperator has two performance parameters, effectiveness and pressure drop, that also have to be selected for the combination of efficiency and cost that creates the best business conditions. Higher effectiveness recuperation requires greater recuperator surface area, which both increases cost and incurs additional pressure drop. Such increased internal pressure drop reduces net power production and consequently increases microturbine cost per kW.

Microturbine performance, in terms of both efficiency and specific power,[2] is highly sensitive to small variations in component performance and internal losses. This is because the high efficiency recuperated cycle processes a much larger amount of air and combustion products flow per kW of net powered delivered than is the case for high-pressure ratio simple-cycle machines. When the net output is the small difference between two large numbers (the

[2] Specific power is power produced by the machine per unit of mass flow through the machine.

compressor and expansion turbine work per unit of mass flow), small losses in component efficiency, internal pressure losses and recuperator effectiveness have large impacts on net efficiency and net power per unit of mass flow.

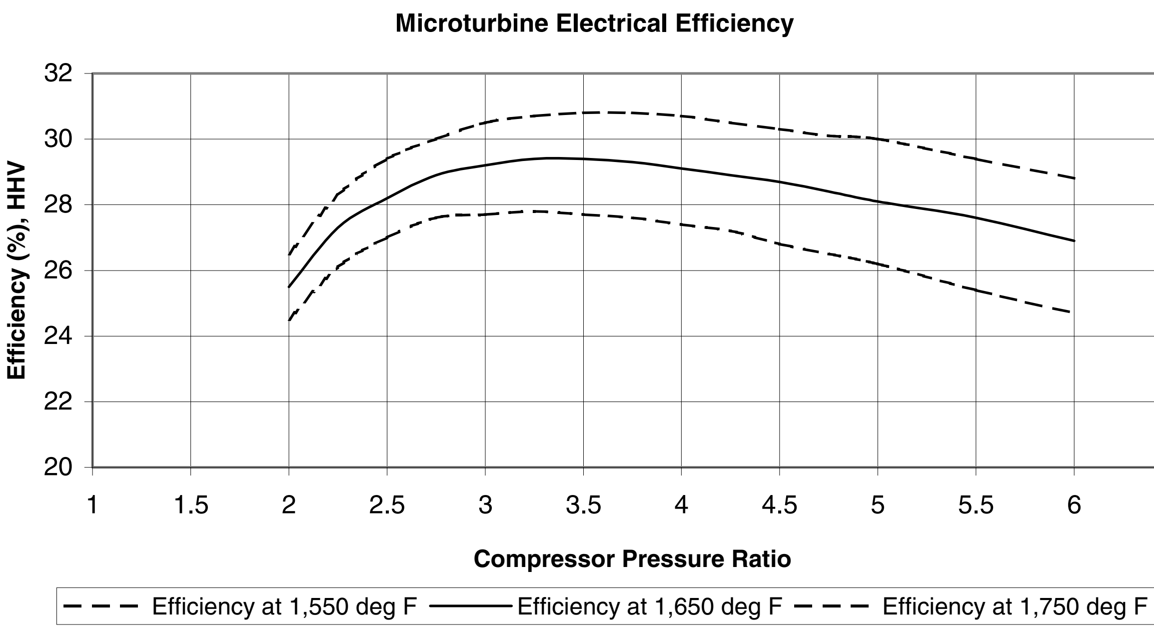
For these reasons, it is advisable to focus on trends and comparisons in considering performance, while relying on manufacturers' guarantees for precise values.

Electrical Efficiency

Figure 2 shows a recuperated microturbine electrical efficiency as a function of microturbine compressor ratio, for a range of turbine firing temperatures from 1,550 to 1,750°F, corresponding to conservative to optimistic turbine material life behavior. The reported efficiency is the gross generator output (without parasitic or conversion losses considered). Often this is at high frequency, so the output must be rectified and inverted to provide 60 Hz AC power. The efficiency loss in such frequency conversion (about 5%, which would lower efficiency from 30% to 28.5%) is not included in these charts. **Figure 2** shows that a broad optimum of performance exists in the pressure ratio range from 3 to 4.

Figure 3 shows microturbine specific power for the same range of firing temperatures and pressure ratios. Higher pressure ratios result in greater specific power. However, practical considerations limit compressor and turbine component tip speed due to centrifugal forces and allowable stresses in economic materials, resulting in compressor pressure ratio limits of 3.5 to 5 in microturbines currently entering the market.

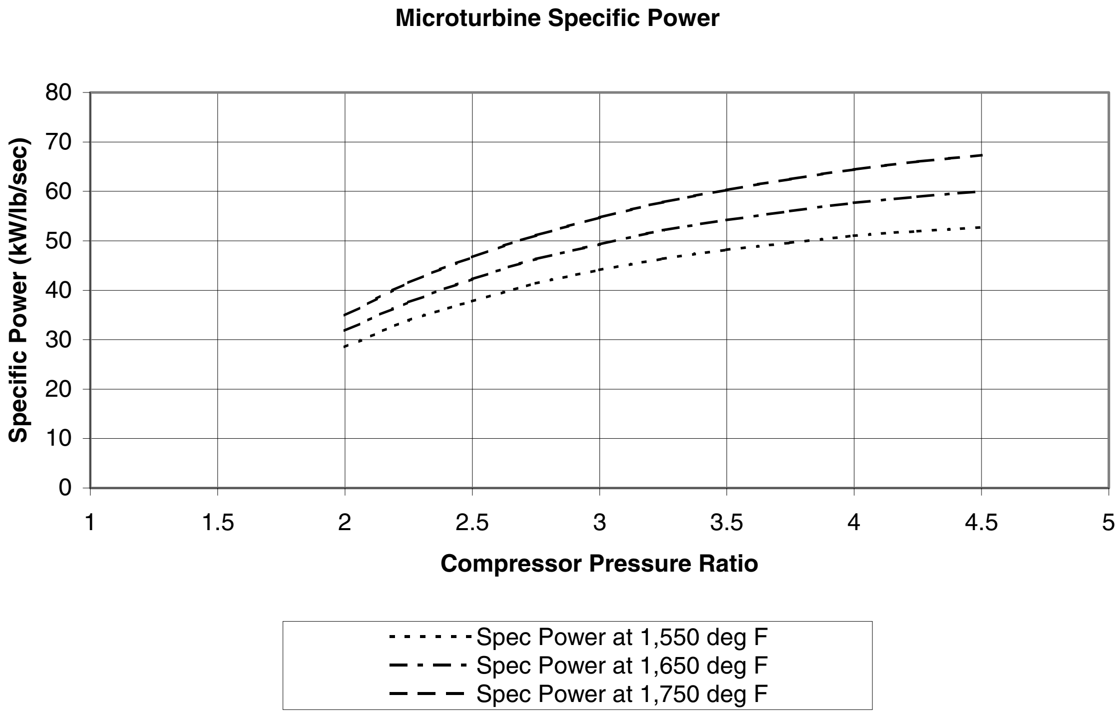
*Figure 2. Microturbine Efficiency as a Function of Compressor Pressure Ratio and Turbine Firing Temperature**



Source: Energy Nexus Group

* Most of the efficiencies quoted in this report are based on higher heating value (HHV), which includes the heat of condensation of the water vapor in the combustion products. In engineering and scientific literature the lower heating value (LHV) is often used, which does not include the heat of condensation of the water vapor in the combustion products). Fuel is sold on a HHV basis. The HHV is greater than the LHV by approximately 10% with natural gas as the fuel (i.e., 50% LHV is equivalent to 45% HHV). HHV efficiencies are about 8% greater for oil (liquid petroleum products) and 5% greater for coal.

Figure 3. Microturbine Specific Power as a Function of Compressor Pressure Ratio and Turbine Firing Temperature



Source: Energy Nexus Group

Table 1 summarizes performance characteristics for typical microturbine CHP systems. The range of 30 to 350 kW represents what is currently or soon to be commercially available. Heat rates and efficiencies shown were taken from manufacturers' specifications and industry publications. Electrical efficiencies are net of parasitic and conversion losses. Available thermal energy is calculated based on manufacturer specifications on turbine exhaust flows and temperatures. CHP thermal recovery estimates are based on producing hot water for process or space heating applications. Total CHP efficiency is the sum of the net electricity generated plus hot water produced for building thermal needs divided by total fuel input to the system. Effective electrical efficiency is a more useful value than overall efficiency to measure fuel savings. Effective electric efficiency assumes that a water heater would otherwise generate the useful thermal output from the CHP system at an 80% thermal efficiency. The theoretical water heating fuel use is subtracted from the total fuel input to calculate the effective electric efficiency of the CHP system.

The data in the table show that electrical efficiency increases as the microturbine becomes larger. As electrical efficiency increases, the absolute quantity of thermal energy available decreases per unit of power output, and the ratio of power to heat for the CHP system increases. A changing ratio of power to heat impacts project economics and may affect the decisions that customers make in terms of CHP acceptance, sizing, and other characteristics.

Table 1. Microturbine CHP - Typical Performance Parameters*

Cost & Performance Characteristics[3]	System 1	System 2	System 3	System 4
Nominal Electricity Capacity (kW)	30 kW	70 kW	100 kW	350 kW
Package Cost (2000 $/kW)[4]	$1,000	$950	$800	$750
Total Installed Cost (2000 $/kW)[5]	$2,516	$2,031	$1,561	$1,339
Electric Heat Rate (Btu/kWh), HHV[6]	14,581	13,540	12,637	11,766
Electrical Efficiency (%), HHV[7]	23.4%	25.2%	27.0%	29.0%
Fuel Input (MMBtu/hr)	0.437	0.948	1.264	4.118
Required Fuel Gas Pressure (psig)	55	55	75	135
CHP Characteristics				
Exhaust Flow (lbs/sec)	0.72	1.40	1.74	5.00
GT Exhaust Temp (degrees F)	500	435	500	600
Heat Exchanger Exhaust Temp (degrees F)	150	130	131	140
Heat Output (MMBtu/hr)	0.218	0.369	0.555	1.987
Heat Output (kW equivalent)	64	108	163	582
Total CHP Efficiency (%), HHV[8]	73%	64%	71%	77%
Power/Heat Ratio[9]	0.47	0.65	0.62	0.60
Net Heat Rate (Btu/kWh)[10]	5,509	6,952	5,703	4,668
Effective Electrical Efficiency (%), HHV[11]	62%	49%	60%	73%

** For typical systems commercially available in 2001 (30-, 70- and 100 kW units) or soon to be available (350 kW model is under development). 30-, 100- and 350 kW systems represented are single-shaft models. 70 kW system represented is a double-shaft model.*

Source: Energy Nexus Group.

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[3] Characteristics presented are representative of "typical" commercially available or soon to be available microturbine systems. Table data are based on: Capstone Model 330 – 30 kW; IR Energy Systems 70LM - 70 kW (two-shaft); Turbec T100 – 100 kW; DTE model currently under development – 350 kW.

[4] Equipment cost only. The cost for all units except for the 30 kW unit includes integral heat recovery water heater. All units include a fuel gas booster compressor.

[5] Installed costs based on CHP system producing hot water from exhaust heat recovery. The 70 kW and 100 kW systems are offered with integral hot water recovery built into the equipment. The 30 kW units are currently built as electric (only) generators and the heat recovery water heater is a separate unit. Other units entering the market are expected to feature built in heat recovery water heaters.

[6] All turbine and engine manufacturers quote heat rates in terms of the lower heating value (LHV) of the fuel. On the other hand, the usable energy content of fuels is typically measured on a higher heating value (HHV) basis. In addition, electric utilities measure power plant heat rates in terms of HHV. For natural gas, the average heat content of natural gas is 1,030 Btu/scf on an HHV basis and 930 Btu/scf on an LHV basis – or about a 10% difference.

[7] Electrical efficiencies are net of parasitic and conversion losses. Fuel gas compressor needs based on 1 psi inlet supply.

[8] Total Efficiency = (net electric generated + net heat produced for thermal needs)/total system fuel input

[9] Power/Heat Ratio = CHP electrical power output (Btu)/ useful heat output (Btu)

[10] Net Heat Rate = (total fuel input to the CHP system - the fuel that would be normally used to generate the same amount of thermal output as the CHP system output assuming an efficiency of 80%)/CHP electric output (kW).

[11] Effective Electrical Efficiency = (CHP electric power output)/(Total fuel into CHP system – total heat recovered/0.8).

Each microturbine manufacturer represented in **Table 1** uses a different recuperator, and each has made individual tradeoffs between cost and performance. Performance involves the extent to which the recuperator effectiveness increases cycle efficiency, the extent to which the recuperator pressure drop decreases cycle power, and the choice of what cycle pressure ratio to use. Consequently, microturbines of different makes will have different CHP efficiencies and different net heat rates chargeable to power.

As shown, microturbines typically require 50 to 80 psig fuel supply pressure. Because microturbines are built with pressure ratios between 3 and 4 to maximize efficiency with a recuperator at modest turbine inlet temperature, the required supply pressure for microturbines is much less than for industrial-size gas turbines with pressure ratios of 7 to 35. Local distribution gas pressures usually range from 30 to 130 psig in feeder lines and from 1 to 50 psig in final distribution lines. Most U.S. businesses that would use a 30, 70, or 100 kW microturbine receive gas at about 0.5 to 1.0 psig. Additionally, most building codes prohibit piping higher-pressure natural gas within the structure. Thus, microturbines in most commercial locations require a fuel gas booster compressor to ensure that fuel pressure is adequate for the gas turbine flow control and combustion systems.

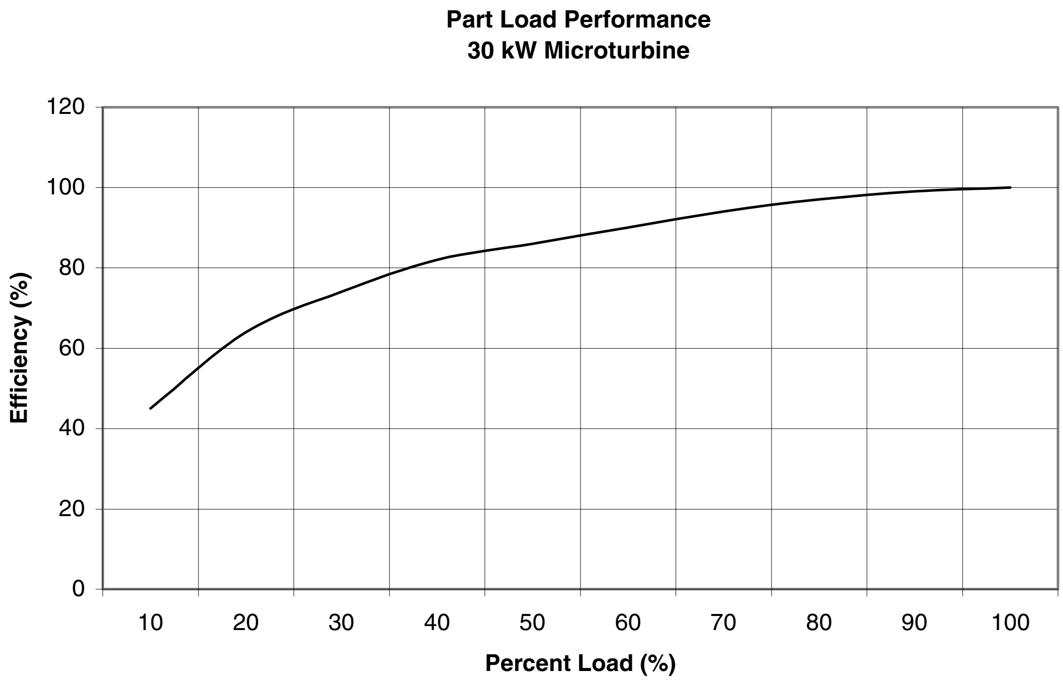
Most microturbine manufacturers offer the equipment package with the fuel gas booster included. It is included in all of the representative systems shown in Table 1. This packaging facilitates the purchase and installation of a microturbine, as the burden of obtaining and installing the booster compressor is no longer placed on the customer. Also, it might result in higher reliability of the booster through standardized design and volume manufacture.

Booster compressors can add from $50 to $100 per kW to a microturbine CHP system's total cost. As well as adding to capital cost, booster compressors lower net power and efficiency so operating cost is slightly higher. Typically, the fuel gas booster requires about 5% of the microturbine output. For example, a single 60 kW unit requires 2.6 kW for the booster, while a booster serving a system of three 30 kW units would require 4.4 kW. Such power loss results in a penalty on efficiency of about 1.5 percentage points. For installations where the unit is located outdoors, the customer can save on cost and operating expense by having the gas utility deliver gas at an adequate pressure and obtaining a system without a fuel gas booster compressor.

Part-Load Performance

When less than full power is required from a microturbine, the output is reduced by a combination of mass flow reduction (achieved by decreasing the compressor speed) and turbine inlet temperature reduction. It takes approximately 15 seconds for a microturbine to go from no-load to full-load conditions. Rapid off-loading (removal of load) will tend to cause the stored energy in the microturbine to overspeed and damage the turbine. In addition to reducing power, this change in operating conditions also reduces efficiency. **Figure 4** shows a sample part-load derate curve for a microturbine.

Figure 4. Microturbine Part Load Power Performance



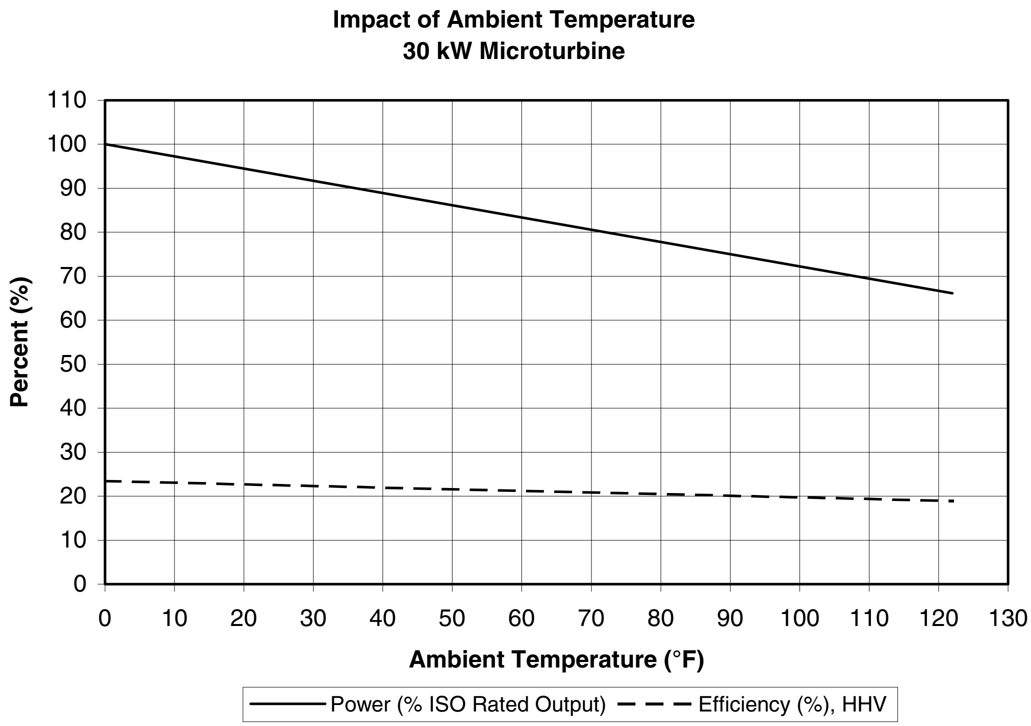
Note: unit represented is a single-shaft, high-speed alternator system.

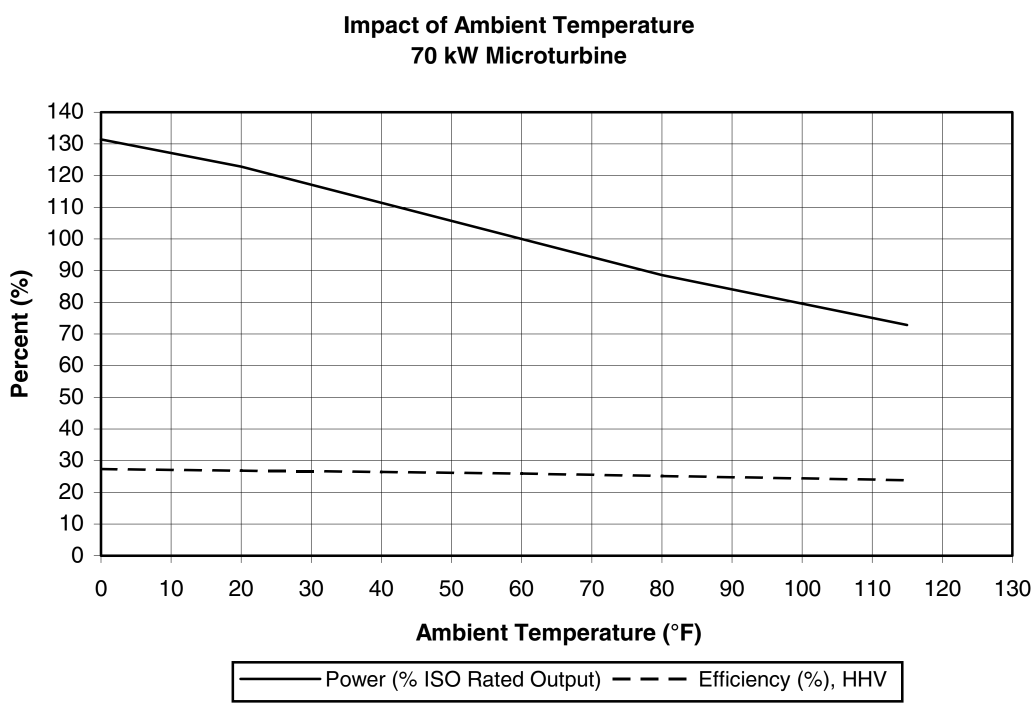
Source: Energy Nexus Group.

Effects of Ambient Conditions on Performance

The ambient conditions under which a microturbine operates have a noticeable effect on both the power output and efficiency. At elevated inlet air temperatures, both the power and efficiency decrease. The power decreases due to the decreased airflow mass rate (since the density of air declines as temperature increases), and the efficiency decreases because the compressor requires more power to compress air of higher temperature. Conversely, the power and efficiency increase with reduced inlet air temperature. **Figure 5** shows the variation in power and efficiency for a microturbine as a function of ambient temperature compared to the reference International Organization for Standards (ISO) condition of sea level and 59°F. The density of air decreases at altitudes above sea level. Consequently, power output decreases. **Figure 6** illustrates the altitude derate.

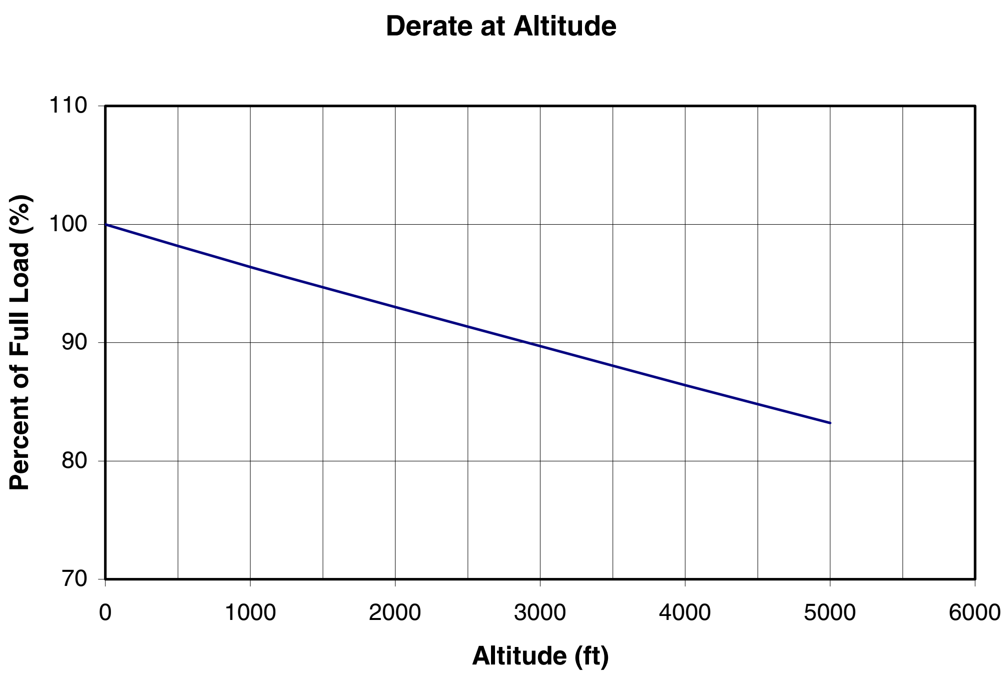
Figure 5. Ambient Temperature Effects on Microturbine Performance





Source: Energy Nexus Group.

Figure 6. Altitude Effects on Microturbine Performance



Source: Energy Nexus Group

Heat Recovery

Effective use of the thermal energy contained in the exhaust gas improves microturbine system economics. Exhaust heat can be recovered and used in a variety of ways, including water heating, space heating, and driving thermally activated equipment such as an absorption chiller or a desiccant dehumidifier.

Microturbine CHP system efficiency is a function of exhaust heat temperature. Recuperator effectiveness strongly influences the microturbine exhaust temperature. Consequently, the various microturbine CHP systems have substantially different CHP efficiency and net heat rate chargeable to power. These variations in CHP efficiency and net heat rate are mostly due to the mechanical design and manufacturing cost of the recuperators and their resulting impact on system cost, rather than being due to differences in system size.

Performance and Efficiency Enhancements

Recuperators

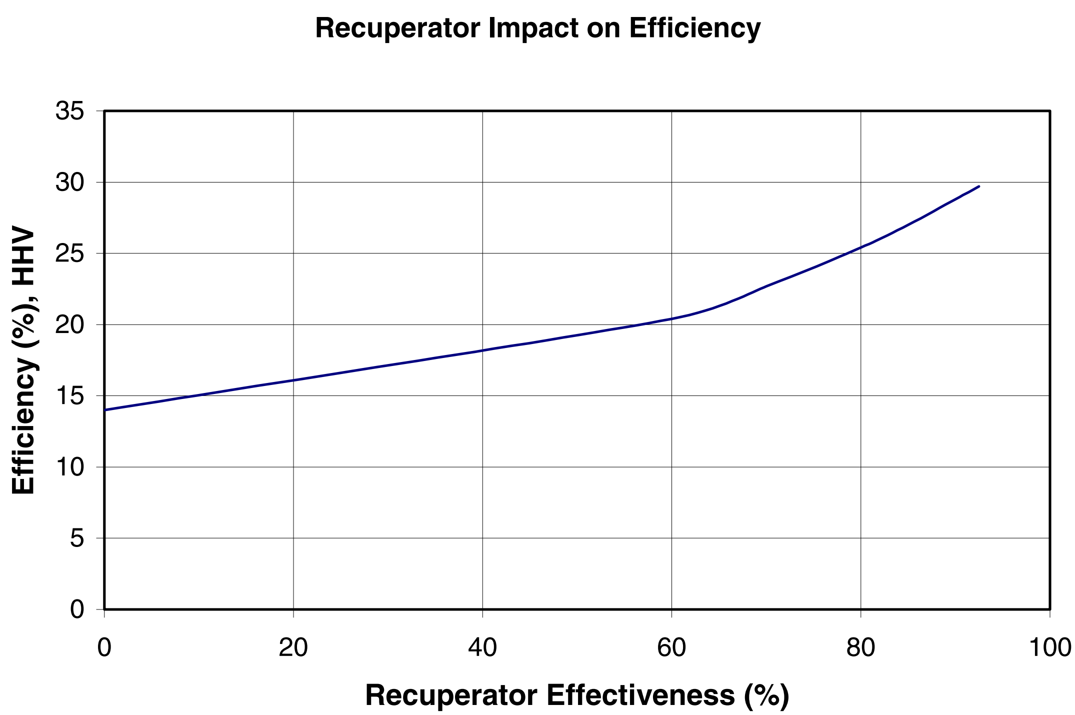
Most microturbines include built in recuperators. The inclusion of a high effectiveness (90%)[12] recuperator essentially doubles the efficiency of a microturbine with a pressure ratio of 3.2, from about 14% to about 29% depending on component details. Without a recuperator, such a machine would be suitable only for emergency, backup, or possibly peaking power operation. With the addition of the recuperator, a microturbine can be suitable for intermediate duty or price-sensitive baseload service.

While recuperators previously in use on industrial gas turbines developed leaks attributable to the consequences of differential thermal expansion accompanying thermal transients, microturbine recuperators have proven quite durable in testing to date. This durability has resulted from using higher strength alloys and higher quality welding along with engineering design to avoid the internal differential expansion that causes internal stresses and leakage. Such practical improvements result in recuperators being of appreciable cost, which detracts from the economic attractiveness of the microturbine. The cost of a recuperator becomes easier to justify as the number of full-power operational hours per year increases.

Incorporation of a recuperator into the microturbine results in pressure losses in the recuperator itself and in the ducting that connects it to other components. Typically, these pressure losses result in 10 to 15% less power being produced by the microturbine, and a corresponding loss of a few points in efficiency. The pressure loss parameter in gas turbines that is the measure of lost power is $\delta p/p$. As $\delta p/p$ increases, the net pressure ratio available for power generation decreases, and hence the power capability of the expansion process diminishes as well. **Figure 7** illustrates the relationship between recuperator effectiveness and microturbine efficiency.

[12] Effectiveness is the technical term in the heat exchanger industry for the ratio of the actual heat transferred to the maximum achievable

Figure 7. Microturbine Efficiency as a Function of Recuperator Effectiveness



Source: Energy Nexus Group.

Firing Temperature

Large turbines (25 to 2,000 lbs/second of mass flow) are usually equipped with internal cooling capability to permit operation with firing temperatures well above those of the metallurgical limit of the best gas turbine alloys. Indeed, progress to higher and higher gas turbine efficiency, via higher firing temperatures, has occurred more through the development and advancement of blade and vane internal cooling technology than through the improvement of the high temperature capabilities of gas turbine alloys.

Unfortunately for microturbine development, the nature of the three dimensional shape of radial inflow turbines has not yet lent itself to the development of a manufacturing method that can produce internal cooling. Consequently, microturbines are limited to firing temperatures within the capabilities of gas turbine alloys. An ongoing program at the U. S. Department of Energy (DOE) Office of Energy Efficiency seeks to apply the technology of ceramic radial inflow turbines (previously advanced for the purpose of developing automotive gas turbines) to microturbines, to increase their efficiency to 36% (HHV). The design and materials technology from the previous efforts are applicable, since the automotive gas turbines were in the same size range, and of the same general geometry, as those used in microturbines.

Inlet Air Cooling

As shown in **Figure 5**, the decreased power and efficiency of microturbines at high ambient temperatures means that microturbine performance is at its lowest at the times power is often in greatest demand and most valued. The use of inlet air cooling can mitigate the decreased power and efficiency resulting from high ambient air temperatures. While inlet air cooling is not a feature on today's microturbines, cooling techniques now entering the market on large gas turbines can be expected to work their way to progressively smaller equipment sizes, and, at some future date, be used with microturbines.

Evaporative cooling, a relatively low capital cost technique, is the most likely to be applied to microturbines. It uses a very fine spray of water directly into the inlet air stream. Evaporation of the water reduces the temperature of the air. Since cooling is limited to the wet bulb air temperature, evaporative cooling is most effective when the wet bulb temperature is appreciably below the dry bulb (ordinary) temperature. In most locales with high daytime dry bulb temperatures, the wet bulb temperature is often 20ºF lower. This affords an opportunity for substantial evaporative cooling. However, evaporative cooling can consume large quantities of water, making it difficult to operate in arid climates.

Refrigeration cooling in microturbines is also technically feasible. In refrigeration cooling, a compression-driven or thermally activated (absorption) refrigeration cycle cools the inlet air through a heat exchanger. The heat exchanger in the inlet air stream causes an additional pressure drop in the air entering the compressor, thereby slightly lowering cycle power and efficiency. However, as the inlet air is now substantially cooler than the ambient air, there is a significant net gain in power and efficiency. Electric motor compression refrigeration requires a substantial parasitic power loss. Thermally activated absorption cooling can use waste heat from the microturbine, reducing the direct parasitic loss. The relative complexity and cost of these approaches, in comparison with evaporative cooling, render them less likely.

Finally, it is also technically feasible to use thermal energy storage systems, typically ice, chilled water, or low-temperature fluids, to cool inlet air. These systems eliminate most parasitic losses from the augmented power capacity. Thermal energy storage is a viable option if on-peak power pricing only occurs a few hours a day. In that case, the shorter time of energy storage discharge and longer time for daily charging allow for a smaller and less expensive thermal energy storage system.

Capital Cost

This section provides typical study estimates for the installed cost of microturbine systems. Two configurations are presented: power-only and CHP producing hot water for use on-site. Equipment-only and installed costs are estimated for the four typical microturbine systems. These are "typical" budgetary price levels. It should also be noted that installed costs can vary significantly depending on the scope of the plant equipment, geographical area, competitive market conditions, special site requirements, emissions control requirements, prevailing labor rates, and whether the system is a new or retrofit application.

Table 2 provides cost estimates for combined heat and power applications, assuming that the CHP system produces hot water. The basic microturbine package consists of the turbogenerator package and power electronics. All of the commercial and near-commercial units offer basic interconnection and paralleling functionality as part of the package cost. All but one of the systems offers an integrated heat exchanger heat recovery system for CHP within the package. All current manufacturers also indicate the package price includes that the gas booster compressor. It should be noted that the package prices cited in the table represent manufacturer quotes or estimates. However, only two of the products have any market history, while the other two are planning to enter the market this year. The manufacturer quotes may not reflect actual cost plus profit today, but may instead represent a *forward pricing* strategy in which early units are sold at a loss to develop the market. The information provided for each sample system is as follows:

- 30 kW – Single unit $1,000/kW, including fuel gas compressor, DC-to-AC inverter, all electronic interconnection hardware, but without the heat recovery heat exchanger. Prices are lower for volume purchases, which are favored. (Capstone: nearly 2,000 units shipped to dealer network.)
- 70 kW – Price of $79,900 includes commissioning and the first year of maintenance (at $0.01/kWh and 4,000 hours/year, equivalent to $40/kW). Built-in heat recovery heat exchanger included in price. Generator is a standard 3,600-rpm AC unit; therefore, there is no need for an inverter. Electrical interconnection and fuel gas booster compressor included. For this comparison, prepaid maintenance and commissioning costs have been backed out from the package price.
- 100 kW – A price of $800/kW is offered to distributors for equipment including heat recovery heat exchanger (built-in), fuel gas booster, DC-to-AC inverter and all interconnection hardware.
- 350 kW – Price target of $910/kW for all equipment, including heat recovery heat exchanger, inverter, fuel gas booster and interconnection, installed. For this comparison, the total package was separated into a package price plus labor.

There is little additional equipment that is required for these integrated systems. A heat recovery system has been added where needed, and additional controls and remote monitoring equipment have been added. The total plant cost consists of total equipment cost plus installation labor and materials (including site work), engineering, project management (including licensing, insurance, commissioning and startup), and financial carrying costs during the 6- to 18-month construction period.

The basic equipment costs represent material on the loading dock, ready to ship. The cost to a customer for installing a microturbine-based CHP system includes a number of other factors that increase the total costs by 70 to 80%. Labor/materials represent the labor cost for the civil, mechanical, and electrical work and materials such as ductwork, piping, and wiring. *Total process capital* is the equipment costs plus installation labor and materials.

A number of other costs are incurred on top of total process capital. These costs are often referred to as *soft costs* because they vary widely by installation, by development channel and by approach to project management. Engineering costs are required to design the system and integrate it functionally with the application's electrical and mechanical systems. In this characterization, environmental permitting fees are included here. Project and construction management also includes general contractor markup and bonding and performance guarantees. Contingency is assumed to be 3% of the total equipment cost in all cases. Up-front, financing costs are also included.

Table 2. Estimated Capital Cost for Microturbine Generators in Grid-Interconnected Combined Heat and Power Application

Cost Component	**System 1**	**System 2**	**System 3**	**System 4**
Nominal Capacity (kW)	30	70	100	350
Costs ($/kW)				
Equipment				
Microturbine	$1,000	$1,030	$800	$750
Gas Booster Compressor	incl.	incl.	incl.	Incl.
Heat Recovery	$225	incl.	incl.	Incl.
Controls/Monitoring	$179	$143	$120	$57
Total Equipment	$1,403	$1,173	$920	$807
Labor/Materials	$429	$286	$200	$160
Total Process Capital	$1,832	$1,459	$1,120	$967
Project and Construction Management	$418	$336	$260	$226
Engineering and Fees	$154	$146	$112	$86
Project Contingency	$72	$58	$45	$38
Project Financing (interest during construction)	$40	$32	$25	$21
Total Plant Cost ($/kW)	$2,516	$2,031	$1,561	$1,339

Source: Energy Nexus Group

Since heat recovery is not required for systems that are power-only, the capital costs are lower. For the units that integrate this equipment into the basic package, the savings will be modest, about a $50/kW reduction in the basic package price. In addition, installation labor and materials costs are reduced because there is no need to connect with the application's thermal system or to connect the heat recovery equipment in the case where it is a separate unit. Power-only systems require less engineering time, as integration is required only with the application's electrical system. Project management and construction fees also tend to be lower because it is a more competitive business than CHP. **Table 3** shows the power-only cost estimates.

Table 3. Estimated Capital Cost for Microturbine Generators in Grid-Interconnected Power-Only Application

Cost Component	**System 1**	**System 2**	**System 3**	**System 4**
Nominal Capacity (kW)	30	70	100	350
Costs ($/kW)				
Equipment				
Microturbine	$1,000	$980	$750	$700
Gas Booster Compressor	$0	$0	$0	$0
Heat Recovery	$0	$0	$0	$0
Controls/Monitoring	$179	$143	$120	$57
Total Equipment	$1,179	$1,123	$870	$757
Labor/Materials	$300	$200	$140	$112
Total Process Capital	$1,479	$1,323	$1,010	$869
Project and Construction Management	$266	$245	$188	$206
Engineering and Fees	$130	$85	$64	$44
Project Contingency	$56	$50	$38	$34
Project Financing (interest during construction)	$31	$27	$21	$18
Total Plant Cost ($/kW)	$1,962	$1,729	$1,320	$1,171

Source: Energy Nexus Group

As an emerging product, the capital costs shown in the preceding two tables represent the cost for the early market entry product, though not the cost of the first units into the market. All of the microturbine developer/manufacturers have cost reduction plans and performance enhancing developments for the mature market product.

Maintenance

Microturbines are still on a learning curve in terms of maintenance, as initial commercial units have seen only two to three years of service so far. With relatively few operating hours logged as a group, the units in the field have not yet yielded enough data to allow much definition in the area of maintenance.

Most manufacturers offer service contracts for specialized maintenance priced at about $0.01/kWh. This includes periodic inspections of the combustor (and associated hot section parts) and the oil bearing in addition to regular air and oil filter replacements. There have been microturbines operating in environments with extremely dusty air that have required frequent air filter changes due to the dust in the air.

A gas microturbine overhaul is needed every 20,000 to 40,000 hours depending on manufacturer, design, and service. A typical overhaul consists of replacing the main shaft with the compressor and turbine attached, and inspecting and if necessary replacing the combustor. At the time of the overhaul, other components are examined to determine if wear has occurred, with replacements made as required. Microturbines are usually operated with at least one on-off cycle per day. There is concern about the effects of this type of operation on component durability.

There is no known difference in maintenance for operation on fuels other than natural gas. However, experience with liquid fuels in industrial gas turbines suggests that liquid fueled combustors require more frequent inspections and maintenance than natural gas fueled combustors.

Fuels

Microturbines have been designed to use natural gas as their primary fuel. However, they are able to operate on a variety of fuels, including:

- Liquefied petroleum gas (LPG) – propane and butane mixtures
- Sour gas – unprocessed natural gas as it comes directly from the gas well
- Biogas – any of the combustible gases produced from biological degradation of organic wastes, such as landfill gas, sewage digester gas, and animal waste digester gas
- Industrial waste gases – flare gases and process off-gases from refineries, chemical plants and steel mill
- Manufactured gases – typically low- and medium-Btu gas produced as products of gasification or pyrolysis processes

Contaminants are a concern with some waste fuels, specifically acid gas components (H_2S, halogen acids, HCN; ammonia; salts and metal-containing compounds; organic halogen-, sulfur-, nitrogen-, and silicon-containing compounds); and oils. In combustion, halogen and sulfur compounds form halogen acids, SO_2, some SO_3 and possibly H_2SO_4 emissions. The acids can also corrode downstream equipment. A substantial fraction of any fuel nitrogen oxidizes into NO_x in combustion. Solid particulates must be kept to low concentrations to prevent corrosion and erosion of components. Various fuel scrubbing, droplet separation, and filtration steps will be required if any fuel contaminant levels exceed manufacturer specifications. Landfill gas in particular often contains chlorine compounds, sulfur compounds, organic acids, and silicon compounds which dictate pretreatment.

Availability

With the small number of units in commercial service, information is not yet sufficient to draw conclusions about reliability and availability of microturbines. The basic design and low number of moving parts hold the potential for systems of high availability; manufacturers have targeted availabilities of 98 to 99%. The use of multiple units or backup units at a site can further increase the availability of the overall facility.

Emissions

Microturbines have the potential for extremely low emissions. All microturbines operating on gaseous fuels feature lean premixed (dry low NO_x, or DLN) combustor technology, which was developed relatively recently in the history of gas turbines and is not universally featured on larger gas turbines.

The primary pollutants from microturbines are oxides of nitrogen (NO_x), carbon monoxide (CO), and unburned hydrocarbons. They also produce a negligible amount of sulfur dioxide (SO_2). Microturbines are designed to achieve the objective of low emissions at full load; emissions are often higher when operating at part load.

The pollutant referred to as NO_x is a mixture of mostly NO and NO_2 in variable composition. In emissions measurement it is reported as parts per million by volume in which both species count equally. NO_x forms by three mechanisms: thermal NO_x, prompt NO_x, and fuel-bound NO_x. The predominant NO_x formation mechanism associated with gas turbines is thermal NO_x. Thermal NO_x is the fixation of atmospheric oxygen and nitrogen, which occurs at high combustion temperatures. Flame temperature and residence time are the primary variables that affect thermal NO_x levels. The rate of thermal NO_x formation increases rapidly with flame temperature. Prompt NO_x forms from early reactions of nitrogen modules in the combustion air and hydrocarbon radicals from the fuel. It forms within the flame and typically is about 1 ppm at 15% O_2, and is usually much smaller than the thermal NO_x formation. Fuel-bound NO_x forms when the fuel contains nitrogen as part of the hydrocarbon structure. Natural gas has negligible chemically bound fuel nitrogen.

Incomplete combustion results in both CO and unburned hydrocarbons. CO emissions result when there is insufficient residence time at high temperature. In gas turbines, the failure to achieve CO burnout may result from combustor wall cooling air. CO emissions are also heavily dependent on operating load. For example, a unit operating under low loads will tend to have incomplete combustion, which will increase the formation of CO. CO is usually regulated to levels below 50 ppm for both health and safety reasons. Achieving such low levels of CO had not been a problem until manufacturers achieved low levels of NO_x, because the techniques used to engineer DLN combustors had a secondary effect of increasing CO emissions.

While not considered a regulated pollutant in the ordinary sense of directly affecting public health, emissions of carbon dioxide (CO_2) are of concern due to its contribution to global warming. Atmospheric warming occurs because solar radiation readily penetrates to the surface of the planet but infrared (thermal) radiation from the surface is absorbed by the CO_2 (and other polyatomic gases such as methane, unburned hydrocarbons, refrigerants, water vapor, and volatile chemicals) in the atmosphere, with resultant increase in temperature of the atmosphere. The amount of CO_2 emitted is a function of both fuel carbon content and system efficiency. The fuel carbon content of natural gas is 34 lbs carbon/MMBtu; oil is 48 lbs carbon/MMBtu; and (ash-free) coal is 66 lbs carbon/MMBtu.

Lean Premixed Combustion

Thermal NO_x formation is a function of both the local temperatures within the flame and residence time. In older technology combustors used in industrial gas turbines, fuel and air were separately injected into the flame zone. Such separate injection resulted in high local temperatures where the fuel and air zones intersected. The focus of combustion improvements of the past decade was to lower flame local hot spot temperature using lean fuel/air mixtures whereby zones of high local temperatures were not created. Lean combustion decreases the fuel/air ratio in the zones where NO_x production occurs so that peak flame temperature is less than the stoichiometric adiabatic flame temperature, therefore suppressing thermal NO_x formation.

All microturbines feature lean pre-mixed combustion systems, also referred to as dry low NO_x or dry low emissions (DLE). Lean premixed combustion pre-mixes the gaseous fuel and compressed air so that there are no local zones of high temperatures, or "hot spots," where high levels of NO_x would form. DLN requires specially designed mixing chambers and mixture inlet zones to avoid flashback of the flame. Optimized application of DLN combustion requires an integrated approach to combustor and turbine design. The DLN combustor is an intrinsic part of the turbine design, and specific combustor designs are developed for each turbine application. Full power NO_x emissions below 9 ppmv @ 15% O_2 have been achieved with lean premixed combustion in microturbines.

Catalytic Combustion

In catalytic combustion, fuels oxidize at lean conditions in the presence of a catalyst. Catalytic combustion is a flameless process, allowing fuel oxidation to occur at temperatures below 1,700°F, where NO_x formation is low. The catalyst is applied to combustor surfaces, which cause the fuel/air mixture to react on the catalyst surface and release its initial thermal energy. The combustion reaction in the remaining volume of the lean premixed gas then goes to completion at design temperature. Data from ongoing long term testing indicates that catalytic combustion exhibits low vibration and acoustic noise, only one-tenth to one-hundredth the levels measured in the same turbine equipped with DLN combustors.

Combustion system and gas turbine developers, along with the U.S. DOE, the California Energy Commission, and other government agencies, are pursuing gas turbine catalytic combustion technology. Past efforts at developing catalytic combustors for gas turbines achieved low, single-digit NO_x ppm levels, but failed to produce combustion systems with suitable operating durability. This was typically due to cycling damage and to the brittle nature of the materials used for catalysts and catalyst support systems. Catalytic combustor developers and gas turbine manufacturers are testing durable catalytic and "partial catalytic" systems that are overcoming the problems of past designs. Catalytic combustors capable of achieving NO_x levels below 3 ppm are in full-scale demonstration and are entering early commercial introduction.[13] As with DLN combustion, optimized catalytic combustion requires an integrated approach to combustor

[13] For example, Kawasaki offers a version of their M1A 13X, 1.4 MW gas turbine with a catalytic combustor with less than 3 ppm NOx guaranteed.

and turbine design. Catalytic combustors must be tailored to the specific operating characteristics and physical layout of each turbine design.

Catalytic combustion may be applied to microturbines as well as industrial and utility turbines. Because of the low emissions from DLN combustors, combined with the low turbine inlet temperatures at which microturbines currently operate, it is not expected that catalytic combustion for microturbines will be pursued in the near term.

Microturbine Emissions Characteristics

Table 4 presents typical emissions for microturbine systems. The data shown reflect manufacturers' guaranteed levels.

Table 4. Microturbine Emissions Characteristics

Emissions Characteristics*	System 1	System 2	System 3	System 4
Nominal Electricity Capacity (kW)	30	70	100	350
Electrical Efficiency, HHV	23%	25%	27%	29%
NO_x, ppmv	9	9	15	9
NO_x, lb/MWh[14]	0.54	0.50	0.80	0.53
CO, ppmv	40	9	15	25
CO, lb/MWh	1.46	0.30	0.49	0.72
THC, ppmv	<9	<9	<10	<10
THC, lb/MWh	<0.19	<0.17	<0.19	<0.19
CO_2, (lb/MWh)	1,928	1,774	1,706	1,529
Carbon, (lb/MWh)	526	484	465	417

Note: Estimates are based on manufacturers' guarantees for typical systems commercially available in 2001 (30-, 70- and 100 kW models). The emissions figures for the 350 kW system under development are manufacturer goals.

[14] Conversion from volumetric emission rate (ppmv at 15% O_2) to output based rate (lbs/MWh) for both NO_x and CO based on conversion multipliers provided by Capstone Turbine Corporation and corrected for differences in efficiency.

Technology Characterization: Reciprocating Engines

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Technology Characterization – Reciprocating Engines

Introduction and Summary

Reciprocating internal combustion engines are a widespread and well-known technology. North American production exceeds 35 million units per year for automobiles, trucks, construction and mining equipment, marine propulsion, lawn care, and a diverse set of power generation applications. A variety of stationary engine products are available for a range of power generation market applications and duty cycles including standby and emergency power, peaking service, intermediate and baseload power, and combined heat and power (CHP). Reciprocating engines are available for power generation applications in sizes ranging from a few kilowatts to over 5 MW.

There are two basic types of reciprocating engines – spark ignition (SI) and compression ignition (CI). Spark ignition engines for power generation use natural gas as the preferred fuel, although they can be set up to run on propane, gasoline, or landfill gas. Compression ignition engines (often called diesel engines) operate on diesel fuel or heavy oil, or they can be set up to run in a dual-fuel configuration that burns primarily natural gas with a small amount of diesel pilot fuel.

Diesel engines have historically been the most popular type of reciprocating engine for both small and large power generation applications. However, in the United States and other industrialized nations, diesel engines are increasingly restricted to emergency standby or limited duty-cycle service because of air emission concerns. Consequently, the natural gas-fueled SI engine is now the engine of choice for the higher-duty-cycle stationary power market (over 500 hr/yr) and is the primary focus of this report.

Current generation natural gas engines offer low first cost, fast start-up, proven reliability when properly maintained, excellent load-following characteristics, and significant heat recovery potential. Electric efficiencies of natural gas engines range from 28% LHV for small stoichiometric engines (<100 kW) to over 40% LHV for large lean burn engines (> 3 MW)[1]. Waste heat recovered from the hot engine exhaust and from the engine cooling systems produces either hot water or low pressure steam for CHP applications. Overall CHP system efficiencies (electricity and useful thermal energy) of 70 to 80% are routinely achieved with natural gas engine systems.

Reciprocating engine technology has improved dramatically over the past three decades, driven by economic and environmental pressures for power density improvements (more output per unit of engine displacement), increased fuel efficiency and reduced emissions. Computer systems have greatly advanced reciprocating engine design and control, accelerating advanced engine

[1] Lower Heating Value. Most of the efficiencies quoted in this report are based on higher heating value (HHV), which includes the heat of condensation of the water vapor in the combustion products. In engineering and scientific literature the lower heating value (LHV – which does not include the heat of condensation of the water vapor in the combustion products) is often used. The HHV is greater than the LHV by approximately 10% with natural gas as the fuel (i.e., 50% LHV is equivalent to 45% HHV). HHV efficiencies are about 8% greater for oil (liquid petroleum products) and 5% for coal.

designs and making possible more precise control and diagnostic monitoring of the engine process. Stationary engine manufacturers and worldwide engine R&D firms continue to drive advanced engine technology, including accelerating the diffusion of technology and concepts from the automotive market to the stationary market.

The emissions signature of natural gas SI engines in particular has improved significantly in the last decade through better design and control of the combustion process and through the use of exhaust catalysts. Advanced lean burn natural gas engines are available that produce NO_x levels as low as 50 ppmv @ 15% O_2 (dry basis).

Applications

Reciprocating engines are well suited to a variety of distributed generation applications. Industrial, commercial, and institutional facilities in the U.S. and Europe for power generation and CHP. Reciprocating engines start quickly, follow load well, have good part-load efficiencies, and generally have high reliabilities. In many cases, multiple reciprocating engine units further increase overall plant capacity and availability. Reciprocating engines have higher electrical efficiencies than gas turbines of comparable size, and thus lower fuel-related operating costs. In addition, the first costs of reciprocating engine gensets are generally lower than gas turbine gensets up to 3-5 MW in size. Reciprocating engine maintenance costs are generally higher than comparable gas turbines, but the maintenance can often be handled by in-house staff or provided by local service organizations.

Potential distributed generation applications for reciprocating engines include standby, peak shaving, grid support, and CHP applications in which hot water, low-pressure steam, or waste-heat-fired absorption chillers are required. Reciprocating engines are also used extensively as direct mechanical drives in applications such as water pumping, air and gas compression and chilling/refrigeration.

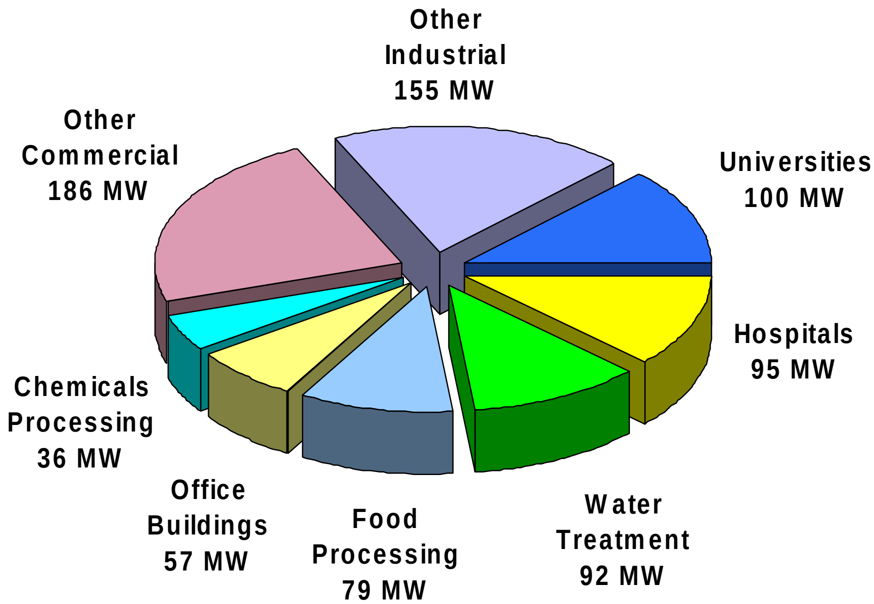
Combined Heat and Power

While the use of reciprocating engines is expected to grow in various distributed generation applications, the most prevalent on-site generation application for natural gas SI engines has traditionally been CHP, and this trend is likely to continue. The economics of natural gas engines in on-site generation applications is enhanced by effective use of the thermal energy contained in the exhaust gas and cooling systems, which generally represents 60 to 70% of the inlet fuel energy.

There are four sources of usable waste heat from a reciprocating engine: exhaust gas, engine jacket cooling water, lube oil cooling water, and turbocharger cooling. Recovered heat is generally in the form of hot water or low pressure steam (<30 psig). The high temperature exhaust can generate medium pressure steam (up to about 150 psig), but the hot exhaust gas contains only about one half of the available thermal energy from a reciprocating engine. Some industrial CHP applications use the engine exhaust gas directly for process drying. Generally, the hot water and low pressure steam produced by reciprocating engine CHP systems is appropriate for low temperature process needs, space heating, potable water heating, and to drive absorption chillers providing cold water, air conditioning, or refrigeration.

There were an estimated 1,055 engine-based CHP systems operating in the United States in 2000 representing over 800 MW of electric capacity.[2] Facility capacities range from 30 kW to 30 MW, with many larger facilities comprised of multiple units. **Figure 1** shows the variety of applications using reciprocating engine CHP**.** Spark ignited engines fueled by natural gas or other gaseous fuels represent 84% of the installed reciprocating engine CHP capacity.

Figure 1. Existing Reciprocating Engine CHP - 801 MW at 1,055 sites



Source: PA Consulting, Energy Nexus Group

Thermal loads most amenable to engine-driven CHP systems in commercial/institutional buildings are space heating and hot water requirements. The simplest thermal load to supply is hot water. **Figure 1** shows the primary applications for CHP in the commercial/institutional sectors are those building types with relatively high and coincident electric and hot water demand such as colleges and universities, hospitals, nursing homes, and lodging. If space heating needs are incorporated, office buildings, certain warehousing, and mercantile/service applications can be economic applications for CHP. Technology development efforts targeted at heat activated cooling/refrigeration and thermally regenerated desiccants expand the application of engine-driven CHP by increasing the thermal energy loads in certain building types. Use of CHP thermal output for absorption cooling and/or desiccant dehumidification could increase the size and improve the economics of CHP systems in existing CHP markets such as schools, lodging, nursing homes, and hospitals. Use of these advanced technologies in applications such as restaurants, supermarkets, and refrigerated warehouses provides a base thermal load that opens these applications to CHP.

[2] PA Consulting *Independent Power Database*, Energy Nexus Group

A typical commercial application for reciprocating engine CHP is a hospital or health care facility with a 1 MW CHP system comprised of multiple 200 to 300 kW natural gas engine gensets. The system design satisfies the baseload electric needs of the facility. Approximately 1.6 MW thermal (MW_{th}) of hot water is recovered from engine exhaust and engine cooling systems to provide space heating and domestic hot water to the facility, and to drive absorption chillers for space conditioning during summer months. Overall efficiency of this type of CHP system can exceed 70%.

Figure 1 shows industry uses engine-driven CHP in a variety of applications where hot water or low-pressure steam is required. A typical industrial application for engine CHP would be a food processing plant with a 2 MW natural gas engine-driven CHP system comprised of multiple 500 to 800 kW engine gensets. The system provides baseload power to the facility and approximately 2.2 MW_{th} low-pressure steam for process heating and washdown. Overall efficiency for a CHP system of this type approaches 75%.

Technology Description

Basic Engine Processes

There are two primary reciprocating engine designs relevant to stationary power generation applications – the spark ignition Otto-cycle engine and the compression ignition Diesel-cycle engine. The essential mechanical components of the Otto-cycle and Diesel-cycle are the same. Both use a cylindrical combustion chamber in which a close fitting piston travels the length of the cylinder. The piston connects to a crankshaft that transforms the linear motion of the piston into the rotary motion of the crankshaft. Most engines have multiple cylinders that power a single crankshaft.

The primary difference between the Otto and Diesel cycles is the method of igniting the fuel. Spark ignition engines (Otto-cycle) use a spark plug to ignite a pre-mixed air fuel mixture introduced into the cylinder. Compression ignition engines (Diesel-cycle) compress the air introduced into the cylinder to a high pressure, raising its temperature to the auto-ignition temperature of the fuel that is injected at high pressure.

Engines are further categorized by crankshaft speed (rpm), operating cycle (2- or 4-stroke), and whether turbocharging is used. Reciprocating engines are also categorized by their original design purpose – automotive, truck, industrial, locomotive, and marine. Hundreds of small-scale stationary power, CHP, irrigation, and chiller applications use automotive engine models. These are generally low-priced engines due to large production volumes. However, unless conservatively rated, these engines have limited durability. Truck engines have the cost benefit of production volume and a reasonably long life (e.g., one million miles). A number of truck engines are available as stationary engines. Engines intended for industrial use are designed for durability and for a wide range of mechanical drive and electric power applications. Their sizes range from 20 kW up to 6 MW, including industrialized truck engines in the 200 to 600 kW range and industrially applied marine and locomotive engines above 1 MW.

Both the spark ignition and the diesel 4-stroke engines most relevant to stationary power generation applications complete a power cycle in four strokes of the piston within the cylinder:

1. Intake stroke – introduction of air (diesel) or air-fuel mixture (spark ignition) into the cylinder.

2. Compression stroke – compression of air or an air-fuel mixture within the cylinder. In diesel engines, the fuel is injected at or near the end of the compression stroke (top dead center or TDC), and ignited by the elevated temperature of the compressed air in the cylinder. In spark ignition engines, the compressed air-fuel mixture is ignited by an ignition source at or near TDC.

3. Power stroke – acceleration of the piston by the expansion of the hot, high pressure combustion gases, and

4. Exhaust stroke – expulsion of combustion products from the cylinder through the exhaust port.

Types of Reciprocating Engines

Natural Gas Spark Ignition Engines – Spark ignition engines use spark plugs, with a high-intensity spark of timed duration, to ignite a compressed fuel-air mixture within the cylinder. Natural gas is the predominant spark ignition engine fuel used in electric generation and CHP applications. Other gaseous and volatile liquid fuels, ranging from landfill gas to propane to gasoline, can be used with the proper fuel system, engine compression ratio and tuning. American manufacturers began to develop large natural gas engines for the burgeoning gas transmission industry after World War II. Smaller engines were developed (or converted from diesel blocks) for gas gathering and other stationary applications as the natural gas infrastructure developed. Natural gas engines for power generation applications are primarily 4-stroke engines available in sizes up to about 5 MW.

Depending on the engine size, one of two ignition techniques ignites the natural gas:

- Open chamber – the spark plug tip is exposed in the combustion chamber of the cylinder, directly igniting the compressed fuel-air mixture. Open chamber ignition is applicable to any engine operating near the stoichiometric air/fuel ratio up to moderately lean mixtures.[3]

- Precombustion chamber – a staged combustion process where the spark plug is housed in a small chamber mounted on the cylinder head. This cylinder charges with a rich mixture of fuel and air, which upon ignition shoots into the main combustion chamber in the cylinder as a high energy torch. This technique provides sufficient ignition energy to light off lean fuel-air mixtures used in large bore engines.[4]

[3] Stoichiometric ratio is the chemically correct ratio of fuel to air for complete combustion, i.e., there is no unused fuel or oxygen after combustion.
[4] Lean mixture is a mixture of fuel and air in which an excess of air is supplied in relation to the amount needed for complete combustion; similarly, a rich mixture is a mixture of fuel and air in which an excess of fuel is supplied in relation to the amount needed for complete combustion.

The simplest natural gas engines operate with natural aspiration of air and fuel into the cylinder (via a carburetor or other mixer) by the suction of the intake stroke. High performance natural gas engines are turbocharged to force more air into the cylinders. Natural gas spark ignition engines operate at modest compression ratios (compared with diesel engines) in the range of 9:1 to 12:1 depending on engine design and turbocharging. Modest compression is required to prevent auto-ignition of the fuel and engine knock, which can cause serious engine damage.[5]

Using high energy ignition technology, lean fuel-air mixtures can be burned in natural gas engines, lowering peak temperatures within the cylinders and resulting in reduced NO_x emissions. The lean burn approach in reciprocating engines is analogous to dry low-NO_x combustors in gas turbines. All major natural gas engine manufacturers offer lean burn, low emission models and are engaged in R&D to further improve their performance.

Natural gas spark ignition engine efficiencies are typically lower than diesel engines because of their lower compression ratios. However, large, high performance lean burn engine efficiencies approach those of diesel engines of the same size. Natural gas engine efficiencies range from about 28% (LHV) for small engines (<50 kW) to 42% (LHV) for the largest high performance, lean burn engines. Lean burn engines tuned for maximum efficiency may produce twice the NO_x emissions as the same engine tuned for minimum NO_x. Tuning for low NO_x typically results in a sacrifice of 1 to 1.5 percentage points in electric generating efficiency from the highest level achievable.

Many natural gas spark ignition engines are derived from diesel engines, i.e., they use the same block, crankshaft, main bearings, camshaft, and connecting rods as the diesel engine. However, natural gas spark ignition engines operate at lower brake mean effective pressure (BMEP) and peak pressure levels to prevent knock.[6] Due to the derating effects from lower BMEP, the spark ignition versions of diesel engines often produce only 60 to 80% of the power output of the parent diesel. Manufacturers often enlarge cylinder bore about 5 to 10% to increase the power, but this is only partial compensation for the derated output. Consequently, the $/kW capital costs of natural gas spark ignition engines are generally higher than the diesel engines from which they were derived. However, by operating at lower cylinder pressure and bearing loads as well as in the cleaner combustion environment of natural gas, spark ignition engines generally offer the benefits of extended component life compared to their diesel parents.

Diesel Engines - Compression ignition diesel are among the most efficient simple-cycle power generation options on the market. Efficiency levels increase with engine size and range from about 30% (HHV) for small high-speed diesels up to 42 to 48% (HHV) for the large bore, slow speed engines. By 2006, it is expected that efficiencies will improve to a maximum of 52% (HHV). High-speed diesel engines (1,200 rpm) are available up to about 4 MW in size. Low speed diesels (60 to 275 rpm) are available as large as 65 MW.

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[5] Knock is produced by explosive auto-ignition of a portion of the fuel in the cylinder due to compression and heating of the gas mixture ahead of the flame front. The term knock and detonation are often used interchangeably.
[6] Brake mean effective pressure (BMEP) can be regarded as the "average" cylinder pressure on the piston during the power stroke and is a measure of the effectiveness of engine power output or mechanical efficiency.

Diesel engines typically require compression ratios of 12:1 to 17:1 to heat the cylinder air to a temperature at which the injected fuel will ignite. The quality of fuel injection significantly affects diesel engine operating characteristics, fuel efficiency, and emissions. Fine atomization and good fuel dispersion by the injectors are essential for rapid ignition, ideal combustion and emissions control. Manufacturers are increasingly moving toward electronically controlled, high-pressure injection systems that provide more precise calibration of fuel delivery and accurate injection timing.

Depending on the engine and fuel quality, diesel engines produce 5 to 20 times the NO_x (on a ppmv basis) of a lean burn natural gas engine. Emergency generators and marine engines often emit over 20 lb NO_x/MWh and present on road engines emit less than 13 lb NO_x/MWh. New diesel engines using low sulfur diesel will achieve rates of approximately 0.65 lb NO_x/MWh. Diesel engines also produce assorted heavy hydrocarbons and particulate emissions. However, diesel engines produce significantly less CO than lean burn gas engines. The NO_x emissions from diesels burning heavy oil are typically 25 to 30% higher than diesels using distillate oil. Common NO_x control techniques include delayed fuel injection, exhaust gas recirculation, water injection, fuel-water emulsification, inlet air cooling, intake air humidification, and compression ratio and/or turbocharger modifications. In addition, an increasing number of larger diesel engines are equipped with selective catalytic reduction and oxidation catalyst systems for post-combustion emissions reduction.

High-speed diesel engines generally require high quality fuel oil with good combustion properties. No. 1 and No. 2 distillate oil comprise the standard diesel fuels. Low sulfur distillate is preferred to minimize SO_2 emissions. High-speed diesels are not suited to burning oil heavier than distillate. Heavy fuel oil requires more time for combustion and the combination of high speed and contaminants in lower quality heavy oils cause excessive wear in high-speed diesel engines. Many medium and low speed diesels designs burn heavier oils including low grade residual oils or Bunker C oils.

Dual Fuel Engines – Dual fuel engines are diesel compression ignition engines predominantly fueled by natural gas with a small percentage of diesel oil as the pilot fuel. The pilot fuel auto-ignites and initiates combustion in the main air-fuel mixture. Pilot fuel percentages can range from 1 to 15% of total fuel input. Dual fuel operation is a combination of Diesel and Otto cycle operation, with reduction in the percentage of pilot fuel used it approaches the Diesel cycle more closely. Most dual fuel engines can be switched back and forth on the fly between dual fuel and 100% diesel operation. In general, because of lower diesel oil usage, NO_x, smoke, and particulate emissions are lower for dual fuel engines than for straight diesel operation—particularly at full load. Particulate emissions reduce in line with the percentage reduction in diesel oil consumption while the level of NO_x reduction depends on combustion characteristics (see **Emissions** section). However, CO and unburned hydrocarbon emissions are often higher, partly because of incomplete combustion.

There are three basic types of dual fuel engines:

<u>Conventional, low pressure gas injection</u> engines typically require about 5 to 10% pilot fuel and may be derated to about 80 to 95% of the rated diesel capacity to avoid detonation. The

turndown ratio of the diesel fuel injection system sets the minimum pilot fuel requirement. Conventional diesel injectors cannot reliably turn down to less than 5 to 6% of the full-load injection rate. Natural gas input is controlled at each cylinder by injecting gas before the air intake valves open. NO_x emissions of conventional dual fuel engines are generally in the 5 to 8 gm/kWh range (compared to lean burn natural gas engines with NO_x emissions in the 0.7 to 2.5 gm/kWh range).

<u>High pressure gas injection</u> engines attempt to reduce derating by injecting natural gas at high pressures (3,600 to 5,100 psig) directly into the main combustion chamber as the pilot fuel is injected. However, the parasitic power for gas compression can be as high as 4 to 7% of the rated power output – partly offsetting the benefit of reduced derating. This technology has not proved particularly popular because of this issue and the additional equipment costs required for gas injection. Pilot fuel consumption is typically 3 to 8% and NO_x emissions are generally in the 5 to 8 gm/kWh range.

<u>Micropilot prechamber</u> engines are similar to spark ignition prechamber engines in that the pilot fuel injected into a prechamber provides a high-energy torch that ignites the lean, compressed fuel air mixture in the cylinder. Leaner mixtures than spark ignition engines are achievable since the energy provided by the diesel-fueled micropilot chamber is higher than that obtained with a spark ignition prechamber. Micropilot dual fuel engines with 1% pilot fuel can operate at or close to the diesel engine's compression ratio and BMEP, so little, if any, derating occurs. In this case the high power density and low $/kW cost advantage of the original diesel engine are retained and engine efficiency at 75 to 100% load is close to that of the 100% diesel engine. NO_x and other emissions are comparable to those of lean burn spark ignition prechamber engines (NO_x emissions in the 0.7 to 2.5 gm/kWh range). These engines must be equipped with conventional diesel fuel injectors in order to operate on 100% diesel.

Several independent developers and engine manufacturers are testing and commercializing dual fuel retrofit kits for converting existing diesel engines to dual fuel operation. The level of sophistication of these kits varies widely and some require major engine modifications. Derating, efficiencies, and emissions also vary widely and have yet to be fully tested or certified. However, dual fuel conversions are unlikely to be as low in emissions as dedicated natural gas engines. In addition, manufacturers may not honor warrantees on an engine retrofitted by an independent third party.

Engine Speed Classifications – Reciprocating engines are classified as high-, medium-, or low-speed. **Table 1** presents the standard speed ranges in each class and the types and sizes of engines available. Engine driven electric generators typically must run at fixed (or synchronous) speeds to maintain a constant 50 or 60 Hertz (Hz) output, setting the engine speed needed within the classifications (i.e., a 60 Hz generator driven by a high speed engine would require engine speeds of 1200, 1800, or 3600 rpm versus a 50 Hz generator which requires engine speeds of 1000, 1500, or 3000 rpm)

Table 1. Reciprocating Engine Types by Speed (Available MW Ratings)

Speed Classification	Engine Speed, rpm	Stoic/ Rich Burn, Spark Ignition[7]	Lean Burn, Spark Ignition	Dual Fuel	Diesel
High Speed	1,000-3,600	0.01 – 1.5 MW	0.15 - 3.0 MW	1.0 - 3.5 MW[8]	0.01 – 3.5 MW
Medium Speed	275-1,000	None	1.0 - 6.0 MW	1.0 – 25 MW	0.5 – 35 MW
Low Speed	58-275	None	None	2.0 – 65 MW	2 – 65 MW

Source: SFA Pacific, Inc.

Engine power output is proportional to engine speed, affording high-speed engines the highest output per unit of displacement (cylinder size) and the highest power density. Consequently, high-speed engines generally have the lowest $/kW production costs of the three types. The cost benefits of high-speed engines must be weighed against other factors. Smaller high-speed engines tend to have lower efficiencies than large bore, lower speed engines due in part to the higher surface area to volume ratio for small cylinders resulting in slightly higher heat losses. In addition, higher speed engines tend to have higher wear rates, resulting in shorter periods between minor and major overhauls. These factors are often less important than capital costs for limited duty cycle applications.

Medium speed stationary power engines are largely derived from marine and locomotive engines. Medium-speed engines are higher in cost, but generally higher in efficiency than high-speed engines. Because of their massive physical size and speed-related power reduction, low-speed engines are increasingly being displaced by medium and high speed engines as the primary choice for stationary power applications.

Load Service Ratings – Reciprocating engine manufacturers typically assign three power ratings to engines depending on the intended load service:

- Standby - continuous full or cycling load for a relatively short duration (usually less than 100 hours) – *maximum power output rating*
- Prime – continuous operation for an unlimited time (except for normal maintenance shutdowns), but with regular variations in load – *80 to 85% of the standby rating*
- Baseload – continuous full-load operation for an unlimited time (except for normal maintenance shutdowns) – *70 to 75% of the standby rating.*

[7] Stoichiometric or rich burn combustion is required for the use of 3-way catalytic converters for emissions control.
[8] Micropilot, prechamber dual fuel engines

Design Characteristics

The features that have made reciprocating engines a leading prime mover for CHP and other distributed generation applications include:

Size range:	Reciprocating engines are available in sizes from 10 kW to over 5 MW.
Thermal output:	Reciprocating engines can produce hot water and low-pressure steam.
Fast start-up:	The fast start-up capability of reciprocating engines allows timely resumption of the system following a maintenance procedure. In peaking or emergency power applications, reciprocating engines can quickly supply electricity on demand.
Black-start capability:	In the event of an electric utility outage, starting reciprocating engines requires minimal auxiliary power. Generally only batteries are required.
Availability:	Reciprocating engines have typically demonstrated availability in excess of 95% in stationary power generation applications.
Part-load operation:	The high part-load efficiency of reciprocating engines ensures economical operation in electric load following applications
Reliability and life:	Reciprocating engines have proven to be reliable power generators given proper maintenance.
Emissions:	Diesel engines have relatively high emissions levels of NO_x and particulates. However, natural gas spark ignition engines have improved emissions profiles and are easier to site.

Performance Characteristics

Electrical Efficiency

Table 2 summarizes performance characteristics for typical commercially available natural gas spark ignition engine CHP systems over a 100 kW to 5 MW size range. This size range covers the majority of the market applications for engine-driven CHP. Heat rates and efficiencies shown were taken from manufacturers' specifications and industry publications. Available thermal energy was calculated from published engine data on engine exhaust temperatures and engine jacket and lube system coolant flows. CHP thermal recovery estimates are based on producing hot water. As shown in the table, 50 to 60% of the waste heat from engine systems is recovered from jacket cooling water and lube oil cooling systems at a temperature too low to produce steam. This feature is generally less critical in commercial/institutional applications where it is more common to have hot water thermal loads. Steam can be produced from the exhaust heat if required (maximum pressure of 150 psig), but if no hot water is needed, the amount of heat recovered from the engine is reduced and total CHP system efficiency drops accordingly.

The data in the table show that electrical efficiency increases as engine size becomes larger. As electrical efficiency increases, the absolute quantity of thermal energy available to produce useful thermal energy decreases per unit of power output, and the ratio of power to heat for the CHP system generally increases. A changing ratio of power to heat impacts project economics and may affect the decisions that customers make in terms of CHP acceptance, sizing, and the desirability of selling power.

*Table 2. Gas Engine CHP - Typical Performance Parameters**

Cost & Performance Characteristics[9]	System 1	System 2	System 3	System 4	System 5
Baseload Electric Capacity (kW)	100	300	800	3,000	5,000
Total Installed Cost (2001 $/kW)[10]	$1,515	$1,200	$1000	$920	$920
Electric Heat Rate (Btu/kWh), HHV[11]	11,147	10,967	10,246	9,492	8,758
Electrical Efficiency (%), HHV	30.6%	31.1%	33.3%	36.0%	39.0%
Engine Speed (rpm)	1,800	1,800	1,200	900	720
Fuel Input (MMBtu/hr)	1.11	3.29	8.20	28.48	43.79
Required Fuel Gas Pressure (psig)	<3	<3	<3	43	65
CHP Characteristics					
Exhaust Flow (1000 lb/hr)	1.0	3.3	10.9	48.4	67.1
Exhaust Temperature (Fahrenheit)	1,060	1,067	869	688	698
Heat Recovered from Exhaust (MMBtu/hr)	0.20	0.82	2.12	5.54	7.16
Heat Recovered from Cooling Jacket (MMBtu/hr)	0.37	0.69	1.09	4.37	6.28
Heat Recovered from Lube System (MMBtu/hr)	0	0	0.29	1.22	1.94
Total Heat Recovered (MMBtu/hr)	0.57	1.51	3.50	11.12	15.38
Total Heat Recovered (kW)	167	443	1,025	3,259	4,508
Form of Recovered Heat	Hot H_20	Hot H_20	Hot H_20	Hot H_20	Hot H_20
Total Efficiency (%)[12]	81%	77%	76%	75%	74%
Power/Heat Ratio[13]	0.60	0.68	0.78	0.92	1.11
Net Heat Rate (Btus/kWh)[14]	4,063	4,687	4,774	4,857	4,914
Effective Electrical Efficiency[15]	0.84	0.73	0.71	0.70	0.69

* *For typical systems commercially available in 2001*

Source: Energy Nexus Group[9]

[9] Characteristics for "typical" commercially available natural gas engine gensets. Data based on: MAN 150 kW – 100 kW; Cummins GSK19G – 300 kW; Caterpillar G3516 LE – 800 kW; Caterpillar G3616 LE – 3 MW; Wartsila 5238 LN - 5 MW; Energy use and exhaust flows normalized to nominal system sizes.

[10] Installed costs based on CHP system producing hot water from exhaust heat recovery (250 °F exhaust from heat recovery heat exchanger), and jacket and lube system cooling

[11] All engine manufacturers quote heat rates in terms of the lower heating value (LHV) of the fuel. However the purchase price of fuels on an energy basis is typically measured on a higher heating value basis (HHV). For natural gas, the average heat content of natural gas is 1,030 Btu/kWh on an HHV basis and 930 Btu/kWh on an LHV basis – or about a 10% difference.

[12] Total CHP Efficiency = (net electric generated + net thermal energy recovered)/total engine fuel input

[13] Power/Heat Ratio = (CHP electric power output (Btus))/useful thermal output (Btus)

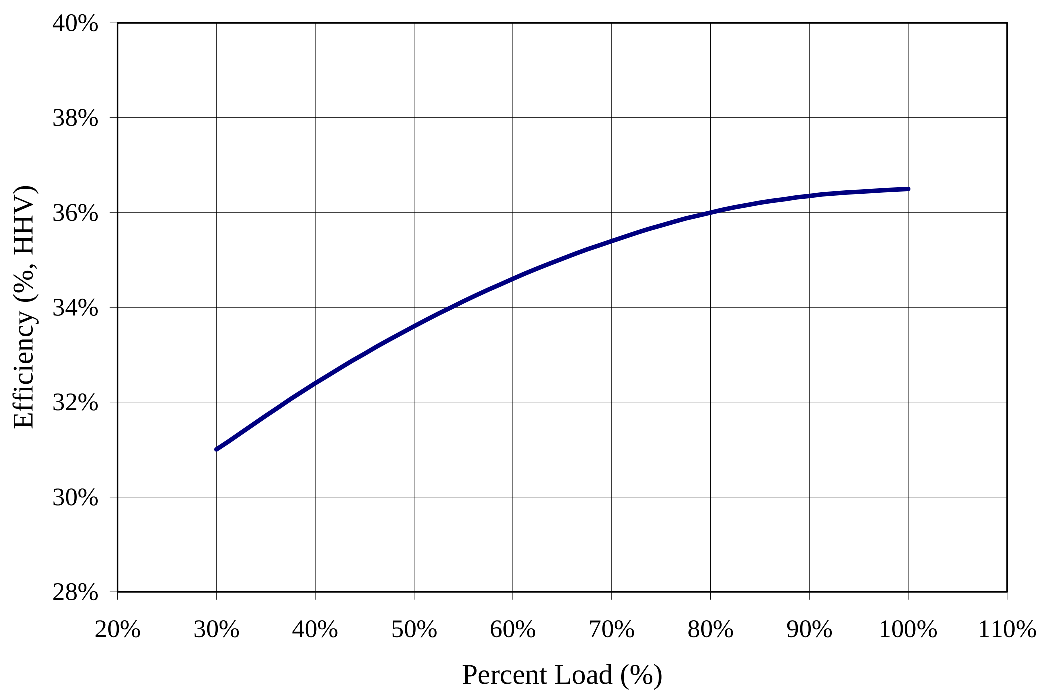
[14] Net Heat Rate = (Total fuel input to the CHP system - the fuel that would be normally used to generate the same amount of thermal output as the CHP system thermal output assuming an efficiency of 80%)/CHP electric output (kW).

[15] Effective Electrical Efficiency = (CHP electric power output)/(Total fuel into CHP system – total heat recovered/0.8); Equivalent to 3,412 Btu/kWh/Net Heat Rate

Part-Load Performance

In power generation and CHP applications, reciprocating engines generally drive synchronous generators at constant speed to produce steady alternating current (AC) power. At reduced loads, the heat rate of spark ignition engines increases and efficiency decreases. **Figure 1** shows the part-load efficiency curve for a typical lean burn natural gas engine. The efficiency at 50% load is approximately 8 to 10% less than full-load efficiency. As the load decreases further, the curve becomes somewhat steeper. While gas engines compare favorably to gas turbines, which typically experience efficiency decreases of 15 to 25% at half-load conditions, multiple engines may be preferable to a single large unit to avoid efficiency penalties where significant load reductions are expected on a regular basis. Diesel engines exhibit even more favorable part-load characteristics than spark ignition engines. The efficiency curve for diesel engines is comparatively flat between 50 and 100% load.

Figure 1. Part-Load Efficiency Performance



Source: Caterpillar, Energy Nexus Group

Effects of Ambient Conditions on Performance

Reciprocating engines are generally rated at ISO conditions of 77°F and 0.987 atmospheres (1 bar) pressure. Like gas turbines, reciprocating engine performance – both output and efficiency – degrades as ambient temperature or site elevation increases. While the effect on gas turbines can be significant, it is less so on engines. Reciprocating engine efficiency and power are reduced by approximately 4% per 1,000 feet of altitude above 1,000 feet, and about 1% for every 10°F above 77°F.

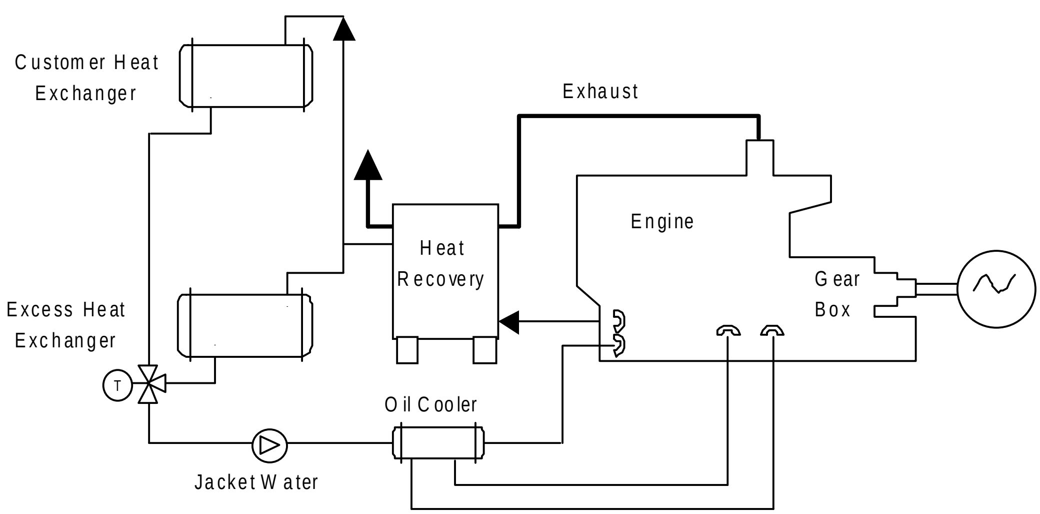
Heat Recovery

The economics of engines in on-site power generation applications often depend on effective use of the thermal energy contained in the exhaust gas and cooling systems, which generally represents 60 to 70% of the inlet fuel energy. Most of the waste heat is available in the engine exhaust and jacket coolant, with smaller amounts recoverable from the lube oil cooler and the turbocharger's intercooler and aftercooler (if so equipped). The most common use of this heat is to generate hot water or low-pressure steam for process use or for space heating, process needs, domestic hot water or absorption cooling. However, the engine exhaust gases can also be used as a source of direct energy for drying or other direct heat processes.

Heat in the engine jacket coolant accounts for up to 30% of the energy input and is capable of producing 200 to 210°F hot water. Some engines, such as those with high pressure or ebullient cooling systems, can operate with water jacket temperatures up to 265°F.   Engine exhaust heat represents from 30 to 50% of the available waste heat. Exhaust temperatures of 850 to 1200°F are typical. By recovering heat in the cooling systems and exhaust, approximately 70 to 80% of the energy of the fuel is effectively utilized to produce both power and useful thermal energy..

Closed-loop cooling systems – **Figure 2** shows the most common method of recovering engine heat, closed-loop cooling system. These systems cool the engine by forced circulation of a coolant through engine passages and an external heat exchanger. An excess heat exchanger transfers engine heat to a cooling tower or radiator when there is excess heat generated. Closed-loop water cooling systems can operate at coolant temperatures from 190 to 250°F. Depending on the engine and CHP system's requirements, the lube oil cooling and turbocharger aftercooling may be either separate or part of the jacket cooling system.

Figure 2. Closed-Loop Heat Recovery System



Ebullient Cooling Systems - Ebullient cooling systems cool the engine by natural circulation of a boiling coolant through the engine. This type of cooling system is typically used in conjunction with exhaust heat recovery for production of low-pressure steam. Cooling water is introduced at the bottom of the engine where the transferred heat begins to boil the coolant generating twophase flow. The formation of bubbles lowers the density of the coolant, causing a natural circulation to the top of the engine.

The coolant at the engine outlet is maintained at saturated steam conditions and is usually limited to 250°F and a maximum of 15 psig. Inlet cooling water is also near saturation conditions and is generally 2 to 3°F below the outlet temperature. The uniform temperature throughout the coolant circuit extends engine life and contributes to improved combustion efficiencies.

Exhaust Heat Recovery – Recovered exhaust heat generates hot water to about 230°F or low-pressure steam (up to 150 psig). To prevent the corrosive effects of condensation in exhaust piping exhaust gas temperatures are generally kept above temperature thresholds, recovering only a portion of the exhaust heat. For this reason, most heat recovery units use a 250 to 350°F exhaust outlet temperature.

Exhaust heat recovery can be independent of the engine cooling system or coupled with it. For example, hot water from the engine cooling can be used as feedwater or feedwater preheat to the exhaust recovery unit. In a typical district heating system, jacket cooling, lube oil cooling, single stage aftercooling and exhaust gas heat recovery are all integrated for steam production.

Performance and Efficiency Enhancements

BMEP and Engine Speed

Engine speed and the BMEP during the power stroke relate to engine power. BMEP is an "average" cylinder pressure on the piston during the power stroke, and is a measure of the effectiveness of engine power output or mechanical efficiency. Engine manufacturers often include BMEP values in their product specifications. Typical BMEP values are as high as 230 psig for large natural gas engines and 350 psig for diesel engines. Corresponding peak combustion pressures are about 1,750 psig and 2,600 psig respectively. High BMEP levels increase power output, improve efficiency, and result in lower specific costs ($/kW).

BMEP can be increased by raising combustion cylinder air pressure through increased turbocharging, improved aftercooling, and reduced pressure losses through improved air passage design. These factors all increase air charge density and raise peak combustion pressures, translating into higher BMEP levels. However, higher BMEP increases thermal and pneumatic stresses within the engine, and proper design and testing is required to ensure continued engine durability and reliability.

Turbocharging

Essentially all modern engines above 300 kW are turbocharged to achieve higher power densities. A turbocharger is a turbine-driven intake air compressor. The hot, high velocity exhaust gases leaving the engine cylinders power the turbine. Large engines typically are equipped with two turbochargers. On a carbureted engine, turbocharging forces more air and fuel into the cylinders, increasing engine output. On a fuel-injected engine, the mass of fuel injected increases in proportion to the increased air input. Cylinder pressure and temperature normally increase as a result of turbocharging, increasing the tendency for detonation for both spark ignition and dual fuel engines and requiring a careful balance between compression ratio and turbocharger boost level. Turbochargers normally boost inlet air pressure on a 3:1 to 4:1 ratio. A wide range of turbocharger designs and models are used. Heat exchangers (called aftercoolers or intercoolers) cool the discharge air from the turbocharger to keep the temperature of the air to the engine under a specified limit.

Capital Cost

This section provides typical study estimates for the installed cost of natural gas spark-ignited, reciprocating engine-driven generators. Two configurations are presented: power-only and CHP. Capital costs (equipment and installation) are estimated for the five typical engine genset systems ranging from 100 kW to 5 MW for each configuration. These are “typical” budgetary price levels; it should also be noted that installed costs vary significantly depending on the scope of the plant equipment, geographical area, competitive market conditions, special site requirements, emissions control requirements, prevailing labor rates, and whether the system is a new or retrofit application.

In general, engine gensets do not show typical economies of scale when costing industrial equipment of different sizes. Smaller genset packages are typically less costly on a unit cost basis ($/kW) than larger gensets. Smaller engines typically run at a higher RPM than larger engines and often are adapted from higher volume production runs from other markets such as automotive or truck engines. These two factors combine to make the engine package costs lower than the larger, slow-speed engines.

The basic genset package consists of the engine connected directly to a generator without a gearbox. In countries where 60 Hz power is required, the genset operates at multiples of 60 – typically 1,800 rpm for smaller engines and 900 or 720 rpm for the large engines. In areas where 50 Hz power is used such as Europe and Japan, the engines run at speeds that are multiples of 50 – typically 1,500 rpm for the small engines. The smaller engines are skid mounted with a basic control system, fuel system, radiator, fan, and starting system. Some smaller packages come with an enclosure, integrated heat recovery system, and basic electric paralleling equipment. The cost of the basic engine genset package plus the costs for added systems needed for the particular application comprise the total equipment cost. The total plant cost consists of total equipment cost plus installation labor and materials (including site work), engineering, project management (including licensing, insurance, commissioning and startup), and financial carrying costs during the 6 to 18 month construction period.

Table 3 provides cost estimates for combined heat and power applications. The CHP system is assumed to produce hot water, although the multi-megawatt size engines are capable of producing low-pressure steam. The heat recovery equipment consists of the exhaust silencer that extracts heat from the exhaust system, process heat exchanger for extracting heat from the engine jacket coolant, circulation pump, control system, and piping. These cost estimates include interconnection and paralleling. The package costs reflect a generic representation of popular engines in each size category. The engines all have low emission, lean-burn technology with the exception of the 100 kW system, which is a rich burn engine that would require a three-way catalyst in most urban installations. The interconnect/electrical costs reflect the costs of paralleling a synchronous generator, though many 100 kW packages available today use induction generators that are simpler and less costly to parallel.[16] Labor/materials represent the labor cost for the civil, mechanical, and electrical work and materials such as ductwork, piping, and wiring. Project and construction management also includes general contractor markup and bonding and performance guarantees. Contingency is assumed to be 3% of the total equipment cost in all cases.

Table 3. Estimated Capital Cost for Typical Gas Engine Generators in Grid Interconnected, Combined Heat and Power Application ($/kW)

Cost Component	System 1	System 2	System 3	System 4	System 5
Nominal Capacity (kW)	100	300	800	3,000	5,000
Costs ($/kW)					
Equipment					
Gen Set Package	$260	$230	$269	$400	$450
Heat Recovery	$205	$179	$89	$65	$40
Interconnect/Electrical	$260	$90	$40	$22	$12
Total Equipment	$725	$499	$398	$487	$502
Labor/Materials	$359	$400	$379	$216	$200
Total Process Capital	$1,084	$899	$777	$703	$702
Project and Construction Management	$235	$158	$121	$95	$95
Engineering and Fees	$129	$81	$45	$41	$41
Project Contingency	$43	$34	$28	$25	$25
Project Financing (interest during construction)	$24	$25	$31	$55	$55
Total Plant Cost ($/kW)	$1,515	$1,197	$1,002	$919	$919

Source: Energy Nexus Group

Maintenance

Maintenance costs vary with type, speed, size and numbers of cylinders of an engine and typically include:

- Maintenance labor.
- Engine parts and material such as oil filters, air filters, spark plugs, gaskets, valves, piston rings, electronic components, etc. and consumables such as oil.
- Minor and major overhauls.

[16] *Reciprocating Engines for Stationary Power Generation: Technology, Products, Players, and Business Issues*, GRI, Chicago, IL and EPRIGEN, Palo Alto, CA: 1999. GRI-99/0271, EPRI TR-113894.

Maintenance can be either done by in-house personnel or contracted out to manufacturers, distributors, or dealers under service contracts. Full maintenance contracts (covering all recommended service) generally cost between 0.7 to 2.0 cents/kWh depending on engine size, speed, and service. Many service contracts now include remote monitoring of engine performance and condition and allow for predictive maintenance. Service contract rates typically are all-inclusive, including the travel time of technicians on service calls.

Recommended service is comprised of routine short interval inspections/adjustments and periodic replacement of engine oil and filter, coolant and spark plugs (typically 500 to 2,000 hours). An oil analysis is part of most preventative maintenance programs to monitor engine wear. A top-end overhaul, generally recommended between 8,000 and 30,000 hours of operation (see **Table 4**), entails a cylinder head and turbocharger rebuild. A major overhaul after 30,000 to 72,000 hours of operation involves piston/liner replacement, crankshaft inspection, bearings, and seals (**Table 4**).

Table 4. Representative Overhaul Intervals for Natural Gas Engines in Baseload Service

	Time Between Overhauls – (Thousand Operating Hours)				
Engine Speed	720 rpm	900 rpm	1,200 rpm	1,500 rpm	1,800 rpm
Minor Overhaul	> 30	15 - 36	24 – 36	10 - 20	8 - 15
Major Overhaul	> 60	40 - 72	48 - 60	30 – 50	30 - 36

Source: SFA Pacific, Inc.

Table 5 presents maintenance costs based on engine manufacturer estimates for service contracts consisting of routine inspections and scheduled overhauls of the engine generator set. Costs are based on 8,000 annual operating hours expressed in terms of annual electricity generation.

Table 5. Typical Natural Gas Engine Maintenance Costs*

Maintenance Costs[17]	System 1	System 2	System 3	System 4	System 5
Electricity Capacity, kW	100	300	800	3,000	5,000
Variable (service contract), $/kWh	0.017	0.012	0.009	0.009	0.009
Variable (consumables), $/kWh	0.00015	0.00015	0.00015	0.00015	0.00015
Fixed, $/kW-yr	10	5	4	1.5	1.1
Fixed, $/kWh @ 8000 hrs/yr	0.00125	0.00063	0.0005	0.00019	0.00014
Total O&M Costs, $/kWh	0.0184	0.0128	0.0097	0.0093	0.0093

* Typical maintenance costs for gas engine gensets 2001

Source: Energy Nexus Group

Fuels

Spark ignition engines operate on a variety of alternative gaseous fuels including:

- Liquefied petroleum gas (LPG) – propane and butane mixtures.
- Sour gas - unprocessed natural gas as it comes directly from the gas well
- Biogas – any of the combustible gases produced from biological degradation of organic wastes, such as landfill gas, sewage digester gas, and animal waste digester gas
- Industrial waste gases – flare gases and process off-gases from refineries, chemical plants and steel mill
- Manufactured gases – typically low- and medium-Btu gas produced as products of gasification or pyrolysis processes.

Factors that impact the operation of a spark ignition engine with alternative gaseous fuels include:

- Volumetric heating value – Since engine fuel is delivered on a volume basis, fuel volume into the engine increases as heating value decreases, requiring engine derating on fuels with very low Btu content. Derating is more pronounced with naturally aspirated engines, and depending on air requirements turbocharging partially or totally compensates.

[17] Maintenance costs presented in **Table 5** are based on 8,000 operating hours expressed in terms of annual electricity generation. Fixed costs are based on an interpolation of manufacturers' estimates. The variable component of the O&M cost represents the inspections and overhaul procedures that are normally conducted by the prime mover original equipment manufacturer through a service agreement usually based on run hours.

- Autoignition characteristics and detonation tendency

- Contaminants that may impact engine component life or engine maintenance, or result in air pollutant emissions that require additional control measures.

- Hydrogen-containing fuels may require special measures (generally if hydrogen content by volume is greater than 5%) because of hydrogen's unique flammability and explosion characteristics.

Table 6 presents representative constituents of some of the alternative gaseous fuels compared to natural gas. Industrial waste and manufactured gases are not included in the table because their compositions vary widely depending on their source. They typically contain significant levels of H_2 and/or CO. Other common constituents are CO_2, water vapor, one or more light hydrocarbons, and H_2S or SO_2.

Table 6. Major Constituents of Gaseous Fuels

	Natural Gas	LPG	Digester Gas	Landfill Gas
Methane, CH_4, (%)	80 – 97	0	35 – 65	40 – 60
Ethane, C_2H_6, (%)	3 – 15	0 – 2	0	0
Propane, C_3H_8, (%)	0 – 3	75 - 97	0	0
Butane,C_4H_{10}, (%)	0 – 0.9	0 - 2	0	0
Higher C_xH_x, (%)	0 – 0.2	0 - 20[18]	0	0
CO_2, (%)	0 – 1.8	0	30 – 40	40 - 60
N_2, (%)	0 – 14	0	1 - 2	0 - 13
H_2, (%)	0 – 0.1	0	0	0
LHV, (Btu/scf)	830 – 1,075	2,500	300 - 600	350 - 550

Source: SFA Pacific, Inc.; North American Combustion Handbook.

Contaminants are a concern with many waste fuels, specifically acid gas components (H_2S, halogen acids, HCN; ammonia; salts and metal-containing compounds; organic halogen-, sulfur-, nitrogen-, and silicon-containing compounds); and oils. In combustion, halogen and sulfur compounds form halogen acids, SO_2, some SO_3 and possibly H_2SO_4 emissions. The acids can also corrode downstream equipment. A substantial fraction of any fuel nitrogen oxidizes into NO_x in combustion. To prevent corrosion and erosion of components solid particulates are kept to low concentrations. Various fuel scrubbing, droplet separation and filtration steps will be required if any fuel contaminant levels exceed manufacturer specifications. Landfill gas in particular often contains chlorine compounds, sulfur compounds, organic acids and silicon compounds, which dictate pretreatment.

[18] High levels of heavier hydrocarbons are found in LPG derived from refinery processing

Once treated and acceptable for use in the engine, emissions performance profiles on alternative fuels are similar to natural gas engine performance. Specifically, the low emissions ratings of lean burn engines can usually be maintained on alternative fuels.

LPG

LPG is composed primarily of propane and/or butane. Propane used in natural gas engines, requires retarding of ignition timing and other appropriate adjustments. LPG often serves as a back-up fuel where there is a possibility of interruption in the natural gas supply. LPG is delivered as a vapor to the engine. LPG's use is limited in high-compression engines because of its relatively low octane number. In general, LPG for engines contains 95% propane by volume with an HHV of 2,500 Btu/scf, and with the remaining 5% lighter than butane. Off-spec LPG may require cooling to condense out larger volumes of butane or heavier hydrocarbons.

High butane content LPG is recommended only for low compression, naturally aspirated engines. Significantly retarded timing avoids detonation.

Field Gas

Field gas often contains more than 5% by volume of heavy ends (butane and heavier), as well as water, salts and H_2S and usually requires some scrubbing before use in natural gas engines. Cooling may be required to reduce the concentrations of butane and heavier components. Field gas usually contains some propane and normally is used in low compression engines (both naturally aspirated and turbocharged). Retarded ignition timing eliminates detonation.

Biogas

Biogases (landfill gas and digester gas) are predominantly mixtures of methane and CO_2 with HHV in the range of 300 to 700 Btu/scf. Landfill gas also contains a variety of contaminants as discussed earlier. Biogases are produced essentially at atmospheric pressure so must be compressed for delivery to the engine. After compression, cooling and scrubbing or filtration are required to remove compressor oil, condensate, and any particulates entrained in the original gas. Scrubbing with a caustic solution may be required if acid gases are present. Because of the additional requirements for raw gas treatment, biogas powered engine facilities are more costly to build and operate than natural gas-based systems.

Industrial Waste Gases

Industrial waste gases that are common reciprocating engine fuels include refinery gases and process off-gases. Refinery gases typically contain components such as H_2, CO, light hydrocarbons, H_2S, and ammonia, as well as CO_2 and N_2. Process off-gases include a variety of compositions. Generally, waste gases are medium- to low-Btu content. Medium-Btu gases generally to not require significant engine derating; low Btu gases usually require derating.

Depending on their origin and contaminants, industrial gases sometimes require pretreatment comparable to that applied to raw landfill gas. Particulates (e.g., catalyst dust), oils, condensable gases, water, C_4+ hydrocarbons and acid gases may all need to be removed. Process offgases are usually available at pressures of several atmospheres or higher, which are generally satisfactory for delivery to an on-site or nearby reciprocating engine facility.

Availability

Reciprocating engines are maintenance intensive, but they can provide high levels of availability, even in high load factor applications. While natural gas engine availabilities vary with engine type, speed, and fuel quality, **Table 7** illustrates typical availability numbers based on a survey of natural gas engine gensets in CHP applications.

Table 7. Availabilities and Outage Rates for Natural Gas Engines

	Gas Engines 80 – 800 kW	Gas Engines >800 kW
Availability Factor (%)	94.5	91.2
Forced Outage Rate (%)	4.7	6.1
Scheduled Outage Rate (%)	2.0	3.5

Source: GRI (Liss, 1999)

The use of multiple units or back-up units at a site can further increase the availability of the overall facility. Some engine manufacturers offer engine exchange programs or other maintenance options that increase the ability to promptly deliver and install replacement units on short notice, typically increasing facility availabilities to greater than 95%.

Emissions

Exhaust emissions are the primary environmental concern with reciprocating engines. The primary pollutants are oxides of nitrogen (NO_x), carbon monoxide (CO), and volatile organic compounds (VOCs – unburned, non-methane hydrocarbons). Other pollutants such as oxides of sulfur (SO_x) and particulate matter (PM) are primarily dependent on the fuel used. The sulfur content of the fuel determines emissions of sulfur compounds, primarily SO_2. Engines operating on natural gas or desulfurized distillate oil emit insignificant levels of SO_x. In general, SO_x emissions are an issue only in large, slow speed diesels firing heavy oils. Particulate matter (PM) can be an important pollutant for engines using liquid fuels. Ash and metallic additives in the fuel contribute to PM in the exhaust.

Nitrogen Oxides (NO_x)

NO_x emissions are usually the primary concern with natural gas engines and are a mixture of (mostly) NO and NO_2 in variable composition. In measurement, NO_x is reported as parts per million by volume in which both species count equally (e.g., ppmv at 15% O_2, dry). Other common units for reporting NO_x in reciprocating engines are gm/hp-hr and gm/kWh, or as an output rate such as lbs/hr. Among engine options, lean burn natural gas engines produce the lowest NO_x emissions and diesel engines produce the highest (**Table 8**).

Table 8. Representative NO_x Emissions from Reciprocating Engines (w/o add on controls)

Engines	Fuel	NO_x (ppmv)	NO_x (gm/kWh)
Diesel Engines (high speed & medium speed)[19]	Distillate	450 – 1,350	7 - 18
Diesel Engines (high speed & medium speed)[20]	Heavy Oil	900 – 1,800	12 – 20
Lean Burn, Spark Ignition Engine[21]	Natural Gas	45 - 150	0.7 – 2.5

Source: SFA Pacific, Inc., Energy Nexus Group

Three mechanism form NO_x: thermal NO_x, prompt NO_x, and fuel-bound NO_x. The predominant NO_x formation mechanism associated with reciprocating engines is thermal NO_x. Thermal NO_x is the fixation of atmospheric oxygen and nitrogen, which occurs at high combustion temperatures. Flame temperature and residence time are the primary variables that affect thermal NO_x levels. The rate of thermal NO_x formation increases rapidly with flame temperature. Early reactions of nitrogen modules in the combustion air and hydrocarbon radicals from the fuel form prompt NO_x. It forms within the flame and typically is approximately 1 ppm at 15% O_2, and is usually much smaller than the thermal NO_x formation. Fuel-bound NO_x forms when the fuel contains nitrogen as part of the hydrocarbon structure. Natural gas has negligible chemically bound fuel nitrogen. Fuel-bound NO_x can be at significant levels with liquid fuels.

The control of peak flame temperature through lean burn conditions has been the primary combustion approach to limiting NO_x formation in gas engines. Diesel engines produce higher combustion temperatures and more NO_x than lean burn gas engines, even though the overall diesel engine air/fuel ratio may be lean. There are three reasons for this: (1) heterogeneous near-stoichiometric combustion; (2) the higher adiabatic flame temperature of distillate fuel; and (3) fuel-bound nitrogen. The diesel fuel is atomized as it is injected and dispersed in the combustion chamber. Combustion largely occurs at near-stoichiometric conditions at the air-droplet and air-fuel vapor interfaces, resulting in maximum temperatures and higher NO_x. In contrast, lean-

[19] Efficiency range: 37 to 44% LHV
[20] Efficiency range: 42 to 48% LHV
[21] Efficiency range: 35 to 42% LHV

premixed homogeneous combustion used in lean burn gas engines results in lower combustion temperatures and lower NO_x production.

For any engine there are generally trade-offs between low NO_x emissions and high efficiency. There are also trade-offs between low NO_x emissions and emissions of the products of incomplete combustion (CO and unburned hydrocarbons). There are three main approaches to these trade-offs that come into play depending on regulations and economics. One approach is to control for lowest NO_x accepting a fuel efficiency penalty and possibly higher CO and hydrocarbon emissions. A second option is finding an optimal balance between emissions and efficiency. A third option is to design for highest efficiency and use post-combustion exhaust treatment.

Carbon Monoxide (CO)

CO and VOCs both result from incomplete combustion. CO emissions result when there is inadequate oxygen or insufficient residence time at high temperature. Cooling at the combustion chamber walls and reaction quenching in the exhaust process also contribute to incomplete combustion and increased CO emissions. Excessively lean conditions can lead to incomplete and unstable combustion and high CO levels.

Unburned Hydrocarbons

Volatile hydrocarbons also called volatile organic compounds (VOCs) can encompass a wide range of compounds, some of which are hazardous air pollutants. These compounds discharge into the atmosphere when some portion of the fuel remains unburned or just partially burned. Some organics are carried over as unreacted trace constituents of the fuel, while others may be pyrolysis products of the heavier hydrocarbons in the gas. Volatile hydrocarbon emissions from reciprocating engines are normally reported as non-methane hydrocarbons (NMHCs). Methane is not a significant precursor to ozone creation and smog formation and is not currently regulated.

Carbon Dioxide (CO_2)

While not considered a pollutant in the ordinary sense of directly affecting health, emissions of carbon dioxide (CO_2) are of concern due to its contribution to global warming. Atmospheric warming occurs since solar radiation readily penetrates to the surface of the planet but infrared (thermal) radiation from the surface is absorbed by the CO_2 (and other polyatomic gases such as methane, unburned hydrocarbons, refrigerants, water vapor, and volatile chemicals) in the atmosphere, with resultant increase in temperature of the atmosphere. The amount of CO_2 emitted is a function of both fuel carbon content and system efficiency. The fuel carbon content of natural gas is 34 lbs carbon/MMBtu; oil is 48 lbs carbon/MMBtu; and (ash-free) coal is 66 lbs carbon/MMBtu.

Emissions Control Options

NO_x control has been the primary focus of emission control research and development in natural gas engines. The following provides a description of the most prominent emission control approaches.

Combustion Process Emissions Control

Control of combustion temperature has been the principal focus of combustion process control in gas engines. Combustion control requires tradeoffs – high temperatures favor complete burn up of the fuel and low residual hydrocarbons and CO, but promote NO_x formation. Lean combustion dilutes the combustion process and reduces combustion temperatures and NO_x formation, and allows a higher compression ratio or peak firing pressures resulting in higher efficiency. However, if the mixture is too lean, misfiring and incomplete combustion occur, increasing CO and VOC emissions.

Lean burn engine technology developed during the 1980s as a direct response to the need for cleaner burning gas engines. As discussed earlier, thermal NO_x formation is a function of both flame temperature and residence time. The focus of lean burn developments was to lower combustion temperature in the cylinder using lean fuel/air mixtures. Lean combustion decreases the fuel/air ratio in the zones where NO_x is produced so that peak flame temperature is less than the stoichiometric adiabatic flame temperature, therefore suppressing thermal NO_x formation. Most lean burn engines use turbocharging to supply excess air to the engine and produce the homogeneous lean fuel-air mixtures. Lean burn engines generally use 50 to 100% excess air (above stoichiometric). The typical emissions rate for lean burn natural gas engines is between 0.5 to 2.0 gm/bhph.

As discussed above, an added performance advantage of lean burn operation is higher output and higher efficiency. Optimized lean burn operation requires sophisticated engine controls to ensure that combustion remains stable and NO_x reduction maximized while minimizing emissions of CO and VOCs. **Table 9** shows data for a large lean burn natural gas engine that illustrates the tradeoffs between NO_x emissions control and efficiency. At the lowest achievable NO_x levels (45 to 50 ppmv), almost 1.5 percentage points are lost on full rated efficiency.

Table 9. NO_x Emissions versus Efficiency Tradeoffs[22]

Engine Characteristics	Low NO_x	High Efficiency
Capacity (MW)	5.2	5.2
Speed (rpm)	720	720
Efficiency, LHV (%)	40.7	42.0
Emissions:		
NO_x (gm/kWh)	0.7	1.4
(ppmv @ 15% O_2)	46	92
CO (gm/kWh)	3.2	2.0
(ppmv @ 15% O_2)	361	227
NMHC (gm/kWh)	0.9	0.6
(ppmv @ 15% O_2)	61	39

Combustion temperature can also be controlled to some extent in reciprocating engines by one or more of the following techniques:

- Delaying combustion by retarding ignition or fuel injection.

- Diluting the fuel-air mixture with exhaust gas recirculation (EGR), which replaces some of the air and contains water vapor that has a relatively high heat capacity and absorbs some of the heat of combustion.

- Introducing liquid water by direct injection or via fuel oil emulsification – evaporation of the water cools the fuel-air mixture charge.

- Reducing the inlet air temperature with a heat exchanger after the turbocharger or via inlet air humidification.

- Modifying valve timing, compression ratio, turbocharging, and the combustion chamber configuration.

Water injection and EGR reduce diesel engine NO_x emissions 30 to 60% from uncontrolled levels. The incorporation of water injection and other techniques to lean burn gas engines is the focus of ongoing R&D efforts with several engine manufacturers and is being pursued as part of the Department of Energy's Advanced Reciprocating Engine Systems (ARES) program. One of the goals of the program is to develop a 45% efficient (HHV) medium sized natural gas engine operating at 0.3 lb NO_x/MWh (0.1 gm NO_x/bhph)

[22] Based on engine manufacturer's data – Wartsila 18V34SG Prechamber Lean Burn Gas Engine.

Post-Combustion Emissions Control

There are several types of catalytic exhaust gas treatment processes that are applicable to various types of reciprocating engines.

Three - Way Catalyst

The catalytic three-way conversion process (TWC) is the basic automotive catalytic converter process that reduces concentrations of all three major criteria pollutants – NO_x, CO and VOCs. The TWC is also called non-selective catalytic reduction (NSCR). NO_x and CO reductions are generally greater than 90%, and VOCs are reduced approximately 80% in a properly controlled TWC system. Because the conversions of NO_x to N_2 and CO and hydrocarbons to CO_2 and H_2O will not take place in an atmosphere with excess oxygen (exhaust gas must contain less than 0.5% O_2), TWCs are only effective with stoichiometric or rich-burning engines. Typical "engine out" NO_x emission rates for a rich burn engine are 10 to 15 gm/bhph. NO_x emissions with TWC control are as low as 0.15 gm/bhph.

Stoichiometric and rich burn engines have significantly lower efficiency than lean burn engines (higher carbon emissions) and only certain sizes (<1.5 MW) and high speeds are available. The TWC system also increases maintenance costs by as much as 25%. TWCs use noble metal catalysts that are vulnerable to poisoning and masking, limiting their use to engines operated with clean fuels – e.g., natural gas and unleaded gasoline. In addition, the engines must use lubricants that do not generate catalyst poisoning compounds and have low concentrations of heavy and base metal additives. Unburned fuel, unburned lube oil, and particulate matter can also foul the catalyst. TWC technology is not applicable to lean burn gas engines or diesels.

Selective Catalytic Reduction (SCR)

This technology selectively reduces NO_x to N_2 in the presence of a reducing agent. NO_x reductions of 80 to 90% are achievable with SCR. Higher reductions are possible with the use of more catalyst or more reducing agent, or both. The two agents used commercially are ammonia (NH_3 in anhydrous liquid form or aqueous solution) and aqueous urea. Urea decomposes in the hot exhaust gas and SCR reactor, releasing ammonia. Approximately 0.9 to 1.0 moles of ammonia is required per mole of NO_x at the SCR reactor inlet in order to achieve an 80 to 90% NO_x reduction.

SCR systems add a significant cost burden to the installation cost and maintenance cost of an engine system, and can severely impact the economic feasibility of smaller engine projects. SCR requires on-site storage of ammonia, a hazardous chemical. In addition, ammonia can "slip" through the process unreacted, contributing to environmental health concerns.

Oxidation Catalysts

Oxidation catalysts generally are precious metal compounds that promote oxidation of CO and hydrocarbons to CO_2 and H_2O in the presence of excess O_2. CO and NMHC conversion levels of 98 to 99% are achievable. Methane conversion may approach 60 to 70%. Oxidation catalysts are now widely used with all types of engines, including diesel engines. They are being used increasingly with lean burn gas engines to reduce their relatively high CO and hydrocarbon emissions.

Lean –NO_x Catalysts

Lean-NO_x catalysts utilize a hydrocarbon reductant (usually the engine fuel) injected upstream of the catalyst to reduce NO_x. While still under development, it appears that NO_x reduction of 80% and both CO and NMHC emissions reductions of 60% may be possible. Long-term testing, however, has raised issues about sustained performance of the catalysts. Current lean-NO_x catalysts are prone to poisoning by both lube oil and fuel sulfur. Both precious metal and base metal catalysts are highly intolerant of sulfur. Fuel use can be significant with this technology – the high NO_x output of diesel engines would require approximately 3% of the engine fuel consumption for the catalyst system.

Gas Engine Emissions Characteristics

Table 10 shows typical emissions for each of the five gas engine systems. The emissions presented assume no exhaust treatment. System 1, 100 kW engine, is a high speed, rich burn engine. Use of a TWC system would reduce NO_x emissions to 0.15 gm/bhph, CO emissions to 0.6 gm/bhph, and VOC emissions to 0.15 gm/bhph. Systems 2 through 5 are all lean burn engines optimized for low emissions. Use of an oxidation catalyst could reduce CO and VOC emissions from these engines by 98 to 99%.

With current commercial technology, highest efficiency and lowest NO_x are not achieved simultaneously. Therefore many manufacturers of lean burn gas engines offer different versions of an engine – a low NO_x version and a high efficiency version – based on different tuning of the engine controls and ignition timing. Achieving highest efficiency operation results in conditions that generally produce twice the NO_x as low NO_x versions (e.g., 1.0 gm/bhph versus 0.5 gm/bhph). Achieving the lowest NO_x typically entails sacrifice of 1 to 2 points in efficiency (e.g., 38% versus 36%). In addition, CO and VOC emissions are higher in engines optimized for minimum NO_x.

*Table 10. Gas Engine Emissions Characteristics <u>Without</u> Exhaust Control Options**

Emissions Characteristics[23]	System 1	System 2	System 3	System 4	System 5
Electricity Capacity (kW)	100	300	800	1000	5000
Electrical Efficiency (HHV)	30.6%	31.1%	33.3%	36.0%	39.0%
Engine Combustion	Rich	Lean	Lean	Lean	Lean
NO_x, (gm/bhph)	15.0	2.0	1.0	0.5	0.5
NO_x, (ppmv @ 15% O_2)	1,100	150	80	44	46
NO_x, (lb/MWh)	44.3	5.91	2.95	1.48	1.48
CO, (gm/bhph)	12.0	1.8	2.6	2.8	2.2
CO, (lb/MWh)	35.4	5.31	7.68	8.27	6.50
VOC, (gm/bhph)	0.7	0.2	1.0	1.4	0.4
VOC, (lb/MWh)	2.07	0.59	2.95	4.13	1.18
CO_2, (lb/MWh)	1,338	1,316	1,166	1,139	1,051
Carbon, (lb/MWh)	365	359	318	311	287

* *For typical systems commercially available in 2001. Emissions estimates for <u>untreated</u> engine exhaust conditions (15% O_2, no TWC, SCR, or other exhaust clean up). Estimates based on typical manufacturers' specifications.*

Source: Energy Nexus Group[23]

[23] Characteristics for "typical" commercially available natural gas engine gensets. Data based on: MAN 150 kW – 100 kW; Cummins GSK19G – 300 kW; Caterpillar G3516 LE – 800 kW; Caterpiller G3616 LE – 1 MW; Wartsila 5238 LN - 3 MW

Technology Characterization: Steam Turbines

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Disclaimer:

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Technology Characterization – Steam Turbines

Introduction and Summary

Steam turbines are one of the most versatile and oldest prime mover technologies still in general production. Power generation using steam turbines has been in use for about 100 years, when they replaced reciprocating steam engines due to their higher efficiencies and lower costs. Conventional steam turbine power plants generate most of the electricity produced in the United States. The capacity of steam turbines can range from 50 kW to several hundred MWs for large utility power plants. Steam turbines are widely used for combined heat and power (CHP) applications.

Unlike gas turbine and reciprocating engine CHP systems where heat is a byproduct of power generation, steam turbines normally generate electricity as a byproduct of heat (steam) generation. A steam turbine is captive to a separate heat source and does not directly convert fuel to electric energy. The energy is transferred from the boiler to the turbine through high-pressure steam that in turn powers the turbine and generator. This separation of functions enables steam turbines to operate with an enormous variety of fuels, from natural gas to solid waste, including all types of coal, wood, wood waste, and agricultural byproducts (sugar cane bagasse, fruit pits, and rice hulls). In CHP applications, steam at lower pressure is extracted from the steam turbine and used directly or is converted to other forms of thermal energy.

Steam turbines offer a wide array of designs and complexity to match the desired application and/or performance specifications. Steam turbines for utility service may have several pressure casings and elaborate design features, all designed to maximize the efficiency of the power plant. For industrial applications, steam turbines are generally of simpler single casing design and less complicated for reliability and cost reasons. CHP can be adapted to both utility and industrial steam turbine designs.

Applications

While steam turbines themselves are competitively priced compared to other prime movers, the costs of complete boiler/steam turbine CHP systems are relatively high on a per kW of capacity basis. This is because of their low power to heat (P/H) ratio, the costs of the boiler, fuel handling, overall steam systems, and the custom nature of most installations. Thus, steam turbines are well suited to medium- and large-scale industrial and institutional applications where inexpensive fuels, such as coal, biomass, various solid wastes and byproducts (e.g., wood chips, etc.), refinery residual oil, and refinery off gases are available. Because of the relatively high cost of the system, high annual capacity factors are required to enable a reasonable recovery of invested capital.

However, a retrofit application of steam turbines into existing boiler/steam systems is often an economic option. A turbine-generator set requires the boiler must be able to support a small increase in demand. In addition, to continue to satisfy thermal demands, the distribution system must be able to accommodate the increased flow rate of lower-Btu steam. In such situations, the decision involves only the added capital cost of the steam turbine, its generator, controls, and electrical interconnection, with the balance of plant already in place. Similarly, many facilities faced with replacement or upgrades of existing boilers and steam systems often consider the addition of steam turbines, especially if steam requirements are relatively large compared to power needs within the facility.

In general, steam turbine applications are driven by balancing lower cost fuel or avoided disposal costs for the waste fuel, with the high capital cost and (hopefully high) annual capacity factor for the steam plant and the combined energy plant-process plant application. For these reasons, steam turbines are not normally direct competitors of gas turbines and reciprocating engines.

Industrial and CHP Applications

The primary locations of steam turbine based CHP systems is industrial processes where solid or waste fuels are readily available for boiler use. In CHP applications, steam extracted from the steam turbine directly feeds into a process or is converted to another form of thermal energy. The turbine may drive an electric generator or equipment such as boiler feedwater pumps, process pumps, air compressors, and refrigeration chillers. Turbines as industrial drivers are usually a single casing machine, either single stage or multistage, condensing or non-condensing depending on steam conditions and the value of the steam. Steam turbines operate at a single speed when driving an electric generator and operate over a speed range when driving a refrigeration compressor. For non-condensing applications, steam exhausted from the turbine is at a pressure and temperature sufficient for the CHP heating application.

There were an estimated 19,062 MW of boiler/steam turbine CHP capacity operating in the United States in 2000 located at over 580 industrial and institutional facilities. **Figure 1** shows the largest amount of capacity is in the chemicals, primary metals, and paper industries. Pulp and paper mills are often an ideal industrial/CHP application for steam turbines. Such facilities operate continuously, have a high demand for steam, and have on-site fuel supply at low, or even negative costs (waste that would have to be otherwise disposed of).

Figure 2 illustrates existing steam turbine CHP capacity by boiler fuel type. While coals fuels much of the installed boiler/steam turbine system base, large amounts of capacity are fueled by wood, waste, and a variety of other fuels.

Figure 1. Existing Boiler/Steam Turbine CHP by Industry

19,062 MW at 582 Sites

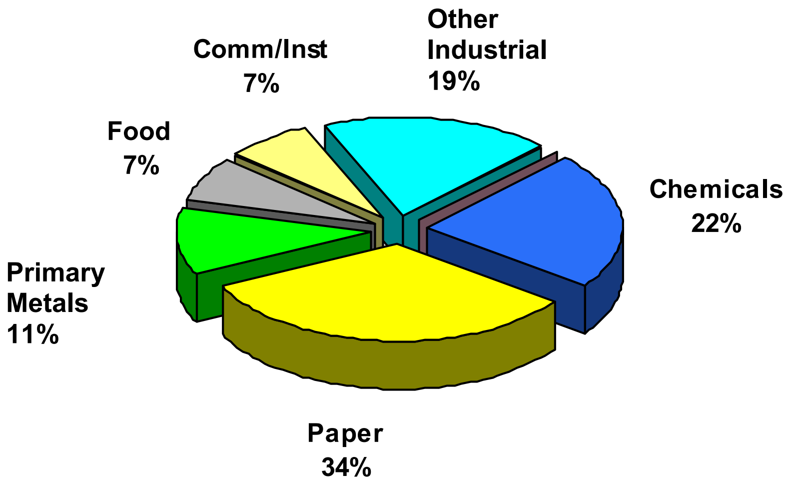
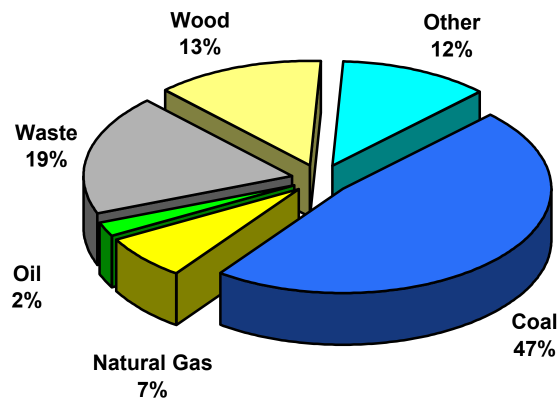


Figure 2. Existing Boiler/Steam Turbine CHP by Boiler Fuel Type

19,062 MW at 582 Sites



Source: Energy Nexus Group/Hagler Bailly.

Combined Cycle Power Plants

The trend in power plant design is the combined cycle, which incorporates a steam turbine in a bottoming cycle with a gas turbine. Steam generated in the heat recovery steam generator (HRSG) of the gas turbine is used to drive a steam turbine to yield additional electricity and improve cycle efficiency. Combined cycle CHP applications use an extraction-condensing type of steam turbine.

District Heating Systems

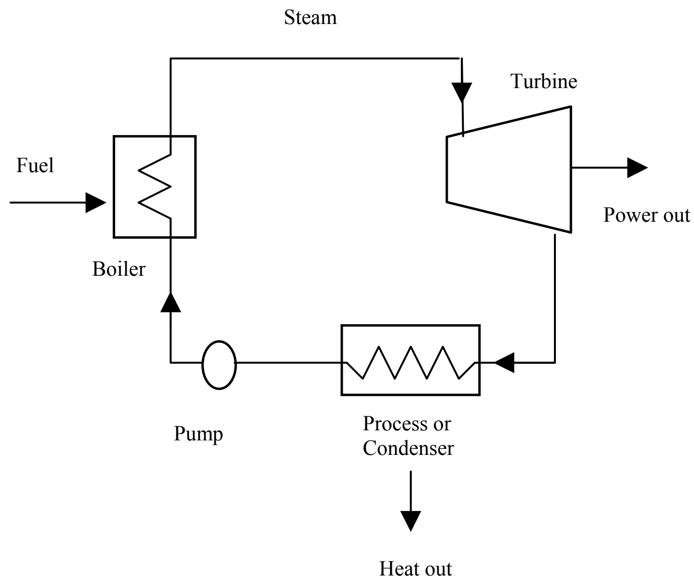
There are many cities and college campuses that have steam district heating systems where adding a steam turbine between the boiler and the distribution system may be an attractive application. Often the boiler is capable of producing moderate-pressure steam but the distribution system needs only low-pressure steam. In these cases, the steam turbine generates electricity using the higher-pressure steam, and discharges low-pressure steam into the distribution system.

Technology Description

Basic Process and Components

The thermodynamic cycle for the steam turbine is the Rankine cycle. The cycle is the basis for conventional power generating stations and consists of a heat source (boiler) that converts water to high-pressure steam. In the steam cycle, water is first pumped to medium to high pressure. It is then heated to the boiling temperature corresponding to the pressure, boiled (heated from liquid to vapor), and then most frequently superheated (heated to a temperature above that of boiling). A multistage turbine expands the pressurized steam to lower pressure and the steam is then exhausted either to a condenser at vacuum conditions or into an intermediate temperature steam distribution system that delivers the steam to the industrial or commercial application. The condensate from the condenser or from the steam utilization system returns to the feedwater pump for continuation of the cycle. **Figure 3** shows the primary components of a boiler/steam turbine system.

Figure 3. Components of a Boiler/Steam Turbine System



The steam turbine itself consists of a stationary set of blades (called nozzles) and a moving set of adjacent blades (called buckets or rotor blades) installed within a casing. The two sets of blades work together such that the steam turns the shaft of the turbine and the connected load. The stationary nozzles accelerate the steam to high velocity by expanding it to lower pressure. A rotating bladed disc changes the direction of the steam flow, thereby creating a force on the blades that, because of the wheeled geometry, manifests itself as torque on the shaft on which the bladed wheel is mounted. The combination of torque and speed is the output power of the turbine.

The internal flow passages of a steam turbine are similar to those of the expansion section of a gas turbine (indeed, gas turbine engineering came directly from steam turbine design around 100 years ago). The main differences are the different gas density, molecular weight, isentropic expansion coefficient, and to a lesser extent viscosity of the two fluids.

Compared to reciprocating steam engines of comparable size, steam turbines rotate at much higher rotational speeds, which contributes to their lower cost per unit of power developed. The absence of inlet and exhaust valves that somewhat throttle (reduce pressure without generating power) and other design features enable steam turbines to be more efficient than reciprocating steam engines. In some steam turbine designs, the blade row accomplishes part of the decrease in pressure and acceleration. These distinctions are known as impulse and reaction turbine designs, respectively. The competitive merits of these designs are the subject of business competition as both designs have sold successfully for well over 75 years.

The connection between the steam supply and the power generation is the steam, and return feedwater, lines. There are numerous options in the steam supply, pressure, temperature and extent, if any, for reheating partially expanded steam. Steam systems vary from low-pressure lines used primarily for space heating and food preparation, to medium pressure used in industrial processes and cogeneration, to high-pressure use in utility power generation. Generally, as the system gets larger the economics favor higher pressures and temperatures with their associated heavier walled boiler tubes and more expensive alloys.

In general, utility applications involve raising steam for the exclusive purpose of power generation. Such systems also use a water-cooled condenser to exhaust the steam from the turbine at the lowest practical pressure. Some utility turbines have dual use, power generation and steam delivery at higher pressure into district heating systems or to neighboring industrial plants at pressure, and consequently do not have condensers. These plants are actually large cogeneration/CHP plants.

Boilers

Steam turbines differ from reciprocating engines and gas turbines in that the fuel is burned in a piece of equipment, the boiler, which is separate from the power generation equipment, the steam turbogenerator. As mentioned previously, this separation of functions enables steam turbines to operate with an enormous variety of fuels.

For sizes up to (approximately) 40 MW, horizontal industrial boilers are built. This enables rail car shipping, with considerable cost savings and improved quality as the cost and quality of factory labor is usually both lower in cost and greater in quality than field labor. Large shop-assembled boilers are typically capable of firing only gas or distillate oil, as there is inadequate residence time for complete combustion of most solid and residual fuels in such designs. Large, field-erected industrial boilers firing solid and residual fuels bear a resemblance to utility boilers except for the actual solid fuel injection. Large boilers usually burn pulverized coal, however intermediate and small boilers burning coal or solid fuel employ various types of solids feeders.

Types of Steam Turbines

The primary type of turbine used for central power generation is the *condensing* turbine. These power-only utility turbines exhaust directly to condensers that maintain vacuum conditions at the discharge of the turbine. An array of tubes, cooled by river, lake, or cooling tower water, condenses the steam into (liquid) water.[1] The cooling water condenses the steam turbine exhaust steam in the condenser creating the condenser vacuum. As a small amount of air leaks into the system when it is below atmospheric pressure, a relatively small compressor removes non-condensable gases from the condenser. Non-condensable gases include both air and a small amount of the corrosion byproduct of the water-iron reaction, hydrogen.

The condensing turbine processes result in maximum power and electrical generation efficiency from the steam supply and boiler fuel. The power output of condensing turbines is sensitive to ambient conditions.[2]

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[1] At 80°F, the vapor pressure of water is 0.51 psia, at 100°F it is 0.95 psia, at 120°F it is 1.69 psia and at 140°F Fahrenheit it is 2.89 psia

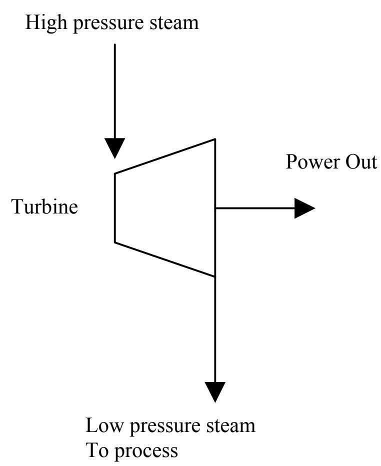
[2] From a reference condition of condensation at 100 degree Fahrenheit, 6.5% less power is obtained from the inlet steam when the temperature at which the steam is condensed is increased (because of higher temperature ambient conditions) to 115°F. Similarly the power output is increased by 9.5% when the condensing temperature is reduced

CHP applications use two types of steam turbines: non-condensing and extraction.

Non-Condensing (Back-pressure) Turbine

Figure 4 shows the non-condensing turbine (also referred to as a back-pressure turbine) exhausts its entire flow of steam to the industrial process or facility steam mains at conditions close to the process heat requirements.

Figure 4. Non-Condensing (Back-Pressure) Steam Turbine



Usually, the steam sent into the mains is not much above saturation temperature.[3] The term "back-pressure" refers to turbines that exhaust steam at atmospheric pressures and above. The specific CHP application establishes the discharge pressure. 50, 150, and 250 psig are the most typical pressure levels for steam distribution systems. District heating systems most often use the lower pressures, and industrial processes use the higher pressures. Industrial processes often include further expansion for mechanical drives, using small steam turbines for driving heavy equipment that runs continuously for long periods. Power generation capability reduces significantly when steam is used at appreciable pressure rather than being expanded to vacuum in a condenser. Discharging steam into a steam distribution system at 150 psig can sacrifice slightly more than half the power that could be generated when the inlet steam conditions are 750 psig and 800°F, typical of small steam turbine systems.

Extraction Turbine

The extraction turbine has opening(s) in its casing for extraction of a portion of the steam at some intermediate pressure before condensing the remaining steam. **Figure 5** illustrates the

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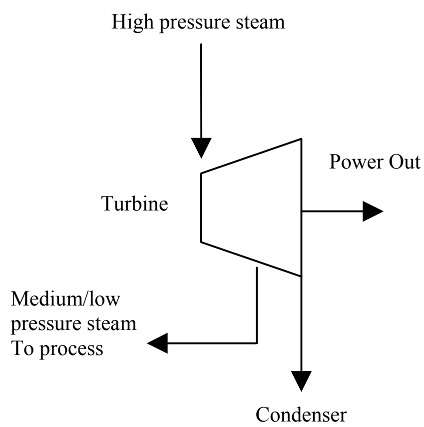
to 80°F. This illustrates the influence of steam turbine discharge pressure on power output and, consequently, net heat rate (and efficiency.)

[3] At 50 psig (65 psia) the condensation temperature is 298°F, at 150 psig (165 psia) the condensation temperature is 366°F, and at 250 psig (265 psia) it is 406°F.

extracted steam may be used for process purposes in a CHP facility or for feedwater heating as is the case in most utility power plants.

Figure 5. Extraction Steam Turbine



The steam extraction pressure may or may not be automatically regulated. Regulated extraction permits more steam to flow through the turbine to generate additional electricity during periods of low thermal demand by the CHP system. In utility type steam turbines, there may be several extraction points, each at a different pressure corresponding to a different temperature. The facility's specific needs for steam and power over time determine the extent to which steam in an extraction turbine is extracted for use in the process.

In large, often complex, industrial plants, additional steam may be admitted (flows into the casing and increases the flow in the steam path) to the steam turbine. Often this happens when using multiple boilers at different pressure, because of their historical existence. These steam turbines are referred to as *admission* turbines. At steam extraction and admission locations there are usually steam flow control valves that add to the steam and control system cost.

Numerous mechanical design features increase efficiency, provide for operation over a range of conditions, simplify manufacture and repair, and achieve other practical purposes. The long history of steam turbine use has resulted in a large inventory of steam turbine stage designs. For example, the division of steam acceleration and change in direction of flow varies between competing turbine manufacturers under the identification of impulse and reaction designs. Manufacturers tailor clients' design requests by varying the flow area in the stages and the extent to which steam is extracted (removed from the flow path between stages) to accommodate the specification of the client.

When the steam expands through a high-pressure ratio, as in utility and large industrial steam systems, the steam can begin to condense in the turbine when the temperature of the steam drops below the saturation temperature at that pressure. If water drops form in the turbine, blade erosion occurs from the drops impact on the blades. At this point in the expansion the steam is sometimes returned to the boiler and reheated to high temperature and then returned to the turbine for further (safe) expansion. In a few large, high pressure, utility steam systems install double reheat systems.

With these choices the designer of the steam supply system and the steam turbine have the challenge of creating a system design which delivers the (seasonally varying) power and steam which presents the most favorable business opportunity to the plant owners.

Between the power (only) output of a condensing steam turbine and the power and steam combination of a back-pressure steam turbine essentially any ratio of power to heat output can be supplied. Back-pressure steam turbines can be obtained with a variety of back pressures, further increasing the variability of the power-to-heat ratio.

Design Characteristics

Custom design:	Steam turbines are designed to match CHP design pressure and temperature requirements and to maximize electric efficiency while providing the desired thermal output.
Thermal output:	Steam turbines are capable of operating over a broad range of steam pressures. Utility steam turbines operate with inlet steam pressures up to 3,500 psig and exhaust vacuum conditions as low as one inch of Hg (absolute). Steam turbines are custom designed to deliver the thermal requirements of the CHP applications through use of backpressure or extraction steam at appropriate pressures and temperatures.
Fuel flexibility:	Steam turbines offer a wide range of fuel flexibility using a variety of fuel sources in the associated boiler or other heat source, including coal, oil, natural gas, wood and waste products.
Reliability and life:	Steam turbine life is extremely long. When properly operated and maintained (including proper control of boiler water chemistry), steam turbines are extremely reliable, only requiring overhauls every several years. They require controlled thermal transients to minimize differential expansion of the parts as the massive casing slowly heats up.
Size range:	Steam turbines are available in sizes from under 100 kW to over 250 MW. In the multi-megawatt size range, industrial and utility steam turbine designations merge, with the same turbine (high-pressure section) able to serve both industrial and small utility applications.
Emissions:	Emissions are dependent upon the fuel used by the boiler or other steam source, boiler furnace combustion section design and operation, and built-in and add-on boiler exhaust cleanup systems.

Performance Characteristics

Electrical Efficiency

The electrical generating efficiency of standard steam turbine power plants varies from a high of 37% HHV[4] for large, electric utility plants designed for the highest practical annual capacity factor, to under 10% HHV for small, simple plants which make electricity as a byproduct of delivering steam to processes or district heating systems.

Steam turbine thermodynamic efficiency (isentropic efficiency) refers to the ratio of power actually generated from the turbine to what would be generated by a perfect turbine with no internal losses using steam at the same inlet conditions and discharging to the same downstream pressure (actual enthalpy drop divided by the isentropic enthalpy drop). Turbine thermodynamic efficiency is not to be confused with electrical generating efficiency, which is the ratio of net power generated to total fuel input to the cycle. Steam turbine thermodynamic efficiency measures how efficiently the turbine extracts power from the steam itself. Multistage (moderate to high-pressure ratio) steam turbines have thermodynamic efficiencies that vary from 65% for small (under 1,000 kW) units to over 90% for large industrial and utility sized units. Small, single stage steam turbines can have efficiencies as low as 50%. When a steam turbine exhausts to a CHP application, the turbine efficiency is not as critical as in a power only condensing mode. The majority of the energy not extracted by the steam turbine satisfies the thermal load. Power only applications waste the exhaust turbine steam energy in condensers.

Table 1 summarizes performance characteristics for typical commercially available steam turbines and for typical boiler/steam CHP systems in the 500 kW to 15 MW size range.

[4] All turbine and engine manufacturers quote heat rates in terms of the lower heating value (LHV) of the fuel. However, the usable energy content of fuels is typically measured on a higher heating value basis (HHV). In addition, electric utilities measure power plant heat rates in terms of HHV. For natural gas, the average heat content of natural gas is 1,030 Btu/scf on an HHV basis and 930 Btu/scf on an LHV basis – or about a 10% difference.

Table 1. Boiler/Steam Turbine CHP System Cost and Performance Characteristics*

Cost & Performance Characteristics[5] Steam Turbine Parameters	System 1	System 2	System 3
Nominal Electricity Capacity (kW)	500	3,000	15,000
Turbine Type	Back Pressure	Back Pressure	Back Pressure
Equipment Cost ($/kW) [6]	540	225	205
Total Installed Cost ($/kW)[7]	918	385	349
Turbine Isentropic Efficiency (%)[8]	50%	70%	80%
Generator/Gearbox Efficiency (%)	94%	94%	97%
Steam Flow (lbs/hr)	21,500	126,000	450,000
Inlet Pressure (psig)	500	600	700
Inlet Temperature (° Fahrenheit)	550	575	650
Outlet Pressure (psig)	50	150	150
Outlet Temperature (° Fahrenheit)	298	366	366
CHP System Parameters			
Boiler Efficiency (%), HHV	80%	80%	80%
CHP Electric Efficiency (%), HHV[9]	6.4%	6.9%	9.3%
Fuel Input (MMBtu/hr)[10]	26.7	147.4	549.0
Steam to Process (MMBtu/hr)	19.6	107.0	386.6
Steam to Process (kW)	5,740	31,352	113,291
Total CHP Efficiency (%), HHV[11]	79.6%	79.5%	79.7%
Power/Heat Ratio[12]	0.09	0.10	0.13
Net Heat Rate (Btu/kWh)[13]	4,515	4,568	4,388
Effective Electrical Efficiency (%), HHV[14]	75.6%	75.1%	77.8%

** For typical systems commercially available in 2002*

[5] Characteristics for "typical" commercially available steam turbine generator systems. Steam turbine data based on information from: TurboSteam, Inc for 500 kW and 3 MW; General Electric for 15 MW turbine.

[6] Equipment cost includes turbine, gearbox, generator, controls, and switchgear; boiler and steam system costs are not included.

[7] Installed costs vary greatly based on site-specific conditions; Installation costs of a "typical" simple installation were estimated to be 70% of the equipment costs.

[8] The Isentropic efficiency of a turbine is a comparison of the actual power output compared to the ideal, or isentropic, output. It is a measure of the effectiveness of extracting work from the expansion process and is used to determine the outlet conditions of the steam from the turbine.

[9] CHP electrical efficiency = Net electricity generated/Total fuel into boiler; A measure of the amount of boiler fuel converted into electricity.

[10] Fuel input based on condensate return at steam outlet pressure and saturation temperature.

[11] Total CHP efficiency = (Net electricity generated+Net steam to process)/Total fuel into boiler

[12] Power/Heat Ratio = CHP electrical power output (Btu)/ useful heat output (Btu)

[13] Net Heat Rate = (total fuel input to the boiler - the fuel that would required to generate the steam to process assuming the same boiler efficiency/steam turbine electric output (kW).

[14] Effective Electrical Efficiency = (Steam turbine electric power output)/(Total fuel into boiler – (steam to process/boiler efficiency)). Equivalent to 3,412 Btu/kWh/Net Heat Rate.

Operating Characteristics

Steam turbines, especially smaller units, leak steam around blade rows and out the end seals. When an end is at a low pressure, as is the case with condensing steam turbines, air can also leak into the system. The leakages cause less power to be produced than expected, and the makeup water has to be treated to avoid boiler and turbine material problems. Air that has leaked in needs to be removed, which is usually done by a compressor removing non-condensable gases from the condenser.

Because of the high pressures used in steam turbines, the casing is quite thick, and consequently steam turbines exhibit large thermal inertia. Steam turbines must be warmed up and cooled down slowly to minimize the differential expansion between the rotating blades and the stationary parts. Large steam turbines can take over ten hours to warm up. While smaller units have more rapid startup times, steam turbines differ appreciably from reciprocating engines, which start up rapidly, and from gas turbines, which can start up in a moderate amount of time and load follow with reasonable rapidity. Steam turbine applications usually operate continuously for extended periods, although the steam fed to the unit and the power delivered may vary (slowly) during such periods of continuous operation.

Process Steam and Performance Tradeoffs

The amount and quality of recovered heat is a function of the entering steam conditions and the design of the steam turbine. Exhaust steam from the turbine is used directly in a process or is converted to other forms of thermal energy, including hot or chilled water. Steam discharged or extracted from a steam turbine can be used in a single- or double effect absorption chiller. The steam turbine can also be used as a mechanical drive for a centrifugal chiller.

CHP System Efficiency

Steam turbine CHP systems generally have low power to heat ratios, typically in the 0.05 to 0.2 range. This is because electricity is a byproduct of heat generation, with the system optimized for steam production. Hence, while steam turbine CHP system electrical efficiency[15] may seem low, it is because the primary objective is to produce large amounts of steam. The effective electrical efficiency[16] of steam turbine systems, however, is generally high, because almost all the energy difference between the high-pressure boiler output and the lower pressure turbine output is converted to electricity. This means that total CHP system efficiencies[17] are generally high and approach the boiler efficiency level. Steam boiler efficiencies range from 70 to 85 % HHV depending on boiler type and age, fuel, duty cycle, application, and steam conditions.

Performance and Efficiency Enhancements

In industrial steam turbine systems, business conditions determine the requirements and relative values of electric power and steam. Plant system engineers then decide the extent of efficiency

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[15] Net power output / total fuel input into the system.
[16] (Steam turbine electric power output)/(Total fuel into boiler – (steam to process/boiler efficiency)).
[17] Net power and steam generated divided by total fuel input.

enhancing options to incorporate in terms of their incremental effects on performance and plant cost, and select appropriate steam turbine inlet and exhaust conditions. Often the steam turbine is going into a system that already exists and is being modified, so that a number of steam system design parameters are already determined by previous decisions, which exist as system hardware characteristics.

As the stack temperature of the boiler exhaust combustion products still contain some heat, tradeoffs occur regarding the extent of investment in heat reclamation equipment for the sake of efficiency improvement. Often the stack exhaust temperature is set at a level where further heat recovery would result in condensation of corrosive chemical species in the stack, with consequential deleterious effects on stack life and safety.

Steam Reheat
Higher pressures and steam reheat increase power generation efficiency in large industrial (and utility) systems. The higher the pressure ratio (the ratio of the steam inlet pressure to the steam exit pressure) across the steam turbine, and the higher the steam inlet temperature, the more power it will produce per unit of mass flow. To avoid condensation the inlet steam temperature is increased to the economic practical limit of materials. This limit is now generally in the range of 800 to 900°F for small industrial steam turbines.

When the economically practical limit of temperature is reached, the expanding steam can reach a condition of temperature and pressure where condensation to (liquid) water begins. Small amounts of water droplets can be tolerated in the last stage of a steam turbine provided that the droplets are not too large or numerous. At pressures higher than that point the steam is returned to the boiler and reheated in temperature and then returned to the expansion steam turbine for further expansion. When returned to the next stage of the turbine, the steam expands without condensation.

Combustion Air Preheating
In large industrial systems, air preheaters recover heat from the boiler exhaust gas stream, and use it to preheat the combustion air, thereby reducing fuel consumption. Boiler combustion air preheaters are large versions of the heat wheels used for the same purpose on industrial furnaces.

Capital Cost

A steam turbine-based CHP plant is a complex process with many interrelated subsystems that must be usually be custom designed. A typical breakdown of installed costs for a steam turbine CHP plant is 25% - boiler, 25% - fuel handling, storage and preparation system, 20% - stack gas cleanup and pollution controls, 15% steam turbine generator, and 20% - field construction and plant engineering. Boiler costs are highly competitive. Typically, the only area in which significant cost reductions can be made when designing a system is in fuel handling/storage/preparation.

In a steam turbine cogeneration plant, especially one burning solid fuel such as biomass, the turbine accounts for a much smaller portion of total system installed costs than is the case with internal combustion engines and industrial gas turbines. Often the solid fuel-handling equipment alone costs as much as 90% of the cost of the steam turbine. The pollution control and electrostatic precipitator cost can reach 80% of the steam turbine cost. A typical coal/wood fired boiler costs more than the steam turbine.[18] The cost of complete solid fuel cogeneration plants varies with many factors, with fuels handling, pollution control equipment and boiler cost all being major cost items. Because of both the size of such plants and the diverse sources of the components, solid fuel cogeneration plants invariably involve extensive system engineering and field labor during construction. Typical complete plant costs run well over $1,000/kW, with little generalization except that for the same fuel and configuration, costs per kW of capacity generally increase as size decreases.

Steam turbine costs exhibit a modest extent of irregularity, as steam turbines are made in sizes with finite steps between the sizes. The cost of the turbine is generally the same for the upper and lower limit of the steam flowing through it, so step-like behavior is sometimes seen in steam turbine prices. Since they come in specific size increments, a steam turbine that is used at the upper end of its range of power capability costs lest per kW generated than one that is used at the lower end of its capability.

Often steam turbines are sold to fit into an existing plant. In some of these applications, the specifications, mass flow, pressure, temperature and backpressure or extraction conditions are not conditions for which large competition exists. These somewhat unique machines are more expensive per kilowatt than are machines for which greater competition exists, for three reasons: 1) a greater amount of custom engineering and manufacturing setup may be required; 2) there is less potential for sales of duplicate or similar units; and 3) there are fewer competitive bidders. The truly competitive products are the "off-the-rack" type machines, while "custom" machines are naturally more expensive.

Steam turbine prices vary greatly with the extent of competition and related manufacturing volumes for units of desired size, inlet and exit steam conditions, rotational speed and standardization of construction. Quoted prices are usually for an assembled steam turbine-electrical generator package. The electrical generator can account for 20% to 40% of the assembly. As the steam turbine/electrical generator package is heavy, due in large part to the heavy walled construction of the high-pressure turbine casing, it requires careful mounting on an appropriate pedestal. The installation and connection to the boiler through high pressure-high temperature steam pipes require engineering and installation expertise. As the high-pressure steam pipes typically vary in temperature by 750°F between cold standby/repair status and full power status, care must be taken in installing a means to accommodate the differential expansion accompanying startup and shutdown. Should the turbine have variable extraction, the cost of the extraction valve and control system adds to the installation.

Small sized steam turbines, below about 2 MW, have a relatively small market, as complete plant cost becomes high enough so that the business venture has much less attractiveness. In these small sizes there is less competition and lower manufacturing volume, so that component costs are not as competitive, the economies of scale in both size and manufacturing volumes disfavor

[18] Spiewak and Weiss, loc. Cit., pages 82 and 95. These figures are for a 32.3 MW multi-fuel fired, 1,250 psig, 900 °F, 50 psig backpressure steam turbine used in an industrial cogeneration plant

such small sizes, and the fraction of total cost due to system engineering and field construction are high.[19]

Boiler combustion produces the steam for a steam turbine and the temperature of the steam is limited by furnace heat transfer design, manufacturing consideration, and boiler tube bundle design. Higher heat fluxes in the boiler enable more compact boilers, with less boiler tube material to be built; however, higher heat fluxes also result in higher boiler tube temperature and the need for the use of a higher grade (adequate strength at higher temperature) boiler tube material. Such engineering economic tradeoffs between temperature (with consequential increases in efficiency) and cost appear throughout the steam plant.

Because of the temperature limitation on boiler tubes, which are exposed to the high temperature and heat flux in the furnace, steam turbine material selection is easier. An often-overlooked component in the steam power system is the steam (safety) stop valve, which is immediately ahead of the steam turbine and is designed to be able to experience the full temperature and pressure of the steam supply. This safety valve is necessary because if the generator electric load were lost (an occasional occurrence), the turbine would rapidly overspeed and destroy itself. Other accidents are possible, supporting the need for the turbine stop valve, which adds significant cost to the system

Maintenance

Steam turbines are rugged units, with operational life often exceeding 50 years. Maintenance is simple, comprised mainly of making sure that all fluids (steam flowing through the turbine and the oil for the bearing) are always clean and at the proper temperature. The oil lubrication system must be clean and at the correct operating temperature and level to maintain proper performance. Other items include inspecting auxiliaries such as lubricating-oil pumps, coolers and oil strainers and checking safety devices such as the operation of overspeed trips.

In order to obtain reliable service, steam turbines require long warmup periods so that there are minimal thermal expansion stresses and wear concerns. Steam turbine maintenance costs are quite low, typically less than $0.004 per kWh. Boilers and any associated solid fuel processing and handling equipment that is part of the boiler/steam turbine plant require their own types of maintenance.

One maintenance issue with steam turbines is solids carry over from the boiler that deposit on turbine nozzles and other internal parts and degrades turbine efficiency and power output. Some of these are water soluble but others are not. Three methods are employed to remove such deposits: 1) manual removal; 2) cracking off deposits by shutting the turbine off and allowing it to cool; and 3) for water soluble deposits, water washing while the turbine is running.

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[19] Data on steam generator costs shows cost increasing with decreasing size, with a 5.25 MW, 900 psig, 850°F, 125 psig backpressure steam turbine/generator costing $285/kW (installed). In that installation the boiler alone, excluding fuel handling and pollution control equipment, cost 150% of the cost of the steam turbine.

Fuels

Industrial boilers operate on a variety of fuels, including wood, coal, natural gas, oils (including residual oil), municipal solid waste, and sludges. The fuel handling, storage, and preparation equipment needed for solid fuels add considerably to the cost of an installation. Thus, such fuels are used only when a high annual capacity factor is expected of the facility, or when the solid material has to be disposed of to avoid an environmental or space occupancy problem.

Availability

Steam turbines generally have 99% plus availability with longer than one year between shutdowns for maintenance and inspections. This high level of availability applies only to the steam turbine, not the boiler or HRSG that is supplying the steam.

Emissions

Emissions associated with a steam turbine are dependent on the source of the steam. Steam turbines can be used with a boiler firing any one or a combination of a large variety of fuel sources, or they can be used with a gas turbine in a combined cycle configuration. Boiler emissions vary depending on fuel type and environmental conditions. Boilers emissions include nitrogen oxide (NO_x), sulfur oxides (SO_x), particulate matter (PM), carbon monoxide (CO), and carbon dioxide (CO_2).

Nitrogen Oxides (NO_x)

The pollutant referred to as NO_x is a mixture of (mostly) nitric oxide (NO) and nitrogen dioxide (NO_2) in variable composition. NO_x forms by three mechanisms: thermal NO_x, prompt NO_x, and fuel-bound NO_x. In industrial boilers, thermal and fuel-bound are the predominant NO_x formation mechanisms. Thermal NO_x, formed when nitrogen and oxygen in the combustion air combine in the flame, comprises the majority of NO_x formed during the combustion of gases and light oils. Fuel-bound NO_x is associated with oil fuels and forms when nitrogen in the fuel and oxygen in the combustion air react.

The most significant factors influencing the level of NO_x emissions from a boiler are the flame temperature and the amount of nitrogen in the fuel. Other factors include excess air level and combustion air temperature.

Sulfur Compounds (SOx)

Emissions of sulfur relate directly to the sulfur content of the fuel, and are not dependent on boiler size or burner design. Sulfur dioxide (SO_2) composes about 95% of the emitted sulfur and with the remaining 5% emitted as sulfur trioxide (SO_3). SO_x are pollutants because they react with water vapor and form sulfuric acid mist, which is extremely corrosive and damaging in its air-, water- and soil-borne forms. Boiler fuels containing sulfur are primarily coal, oil, and some types of waste.

Particulate Matter (PM)

PM emissions are largely dependent on the grade of boiler fuel, and consist of many different compounds, including nitrates, sulfates, carbons, oxides and other uncombusted fuel elements. PM levels from natural gas are significantly lower than those of oils, and distillate oils much lower than residual oils. For industrial and commercial boilers, the most effective method of PM control is use of higher-grade fuel, and ensuring proper burner setup, adjustment and maintenance.

Carbon Monoxide (CO)

CO forms during combustion when carbon in the fuel oxidizes incompletely, ending up as CO instead of CO_2. Older boilers generally have higher levels of CO than new equipment because older burner designs do not have CO controls. Poor burner design or firing conditions are responsible for high levels of CO boiler emissions. Proper burner maintenance or equipment upgrades, or using an oxygen control package, can control CO emissions successfully.

Carbon Dioxide (CO_2)

While not considered a regulated pollutant in the ordinary sense of directly affecting public health, emissions of carbon dioxide are of concern due to its contribution to global warming. Atmospheric warming occurs because solar radiation readily penetrates to the surface of the planet but infrared (thermal) radiation from the surface is absorbed by the CO_2 (and other polyatomic gases such as water vapor, methane, unburned hydrocarbons, refrigerants and volatile chemicals) in the atmosphere, with resultant increase in temperature of the atmosphere. The amount of CO_2 emitted is a function of both fuel carbon content and system efficiency. The fuel carbon content of natural gas is 34 lbs carbon/MMBtu; oil is 48 lbs carbon/MMBtu; and (ash-free) coal is 66 lbs carbon/MMBtu.

Typical Emissions

Table 2 below illustrates typical emissions of NO_x, PM and CO for boilers by size of steam turbine system and by fuel type.

Table 2. Typical Boiler Emissions Ranges

Boiler Fuel	System 1 500 kW			Systems 2 and 3 3 MW / 15 MW		
	NO_x	CO	PM	NO_x	CO	PM
Coal (lbs/MMBtu)	N/A	N/A	N/A	0.20-1.24	0.02-0.7	
Wood (lbs/MMBtu)	0.22-0.49	0.6	0.33-0.56	0.22-0.49	0.06	0.33-0.56
Fuel Oil (lbs/MMBtu)	0.15-0.37	0.03	0.01-0.08	0.07-0.31	0.03	0.01-0.08
Natural Gas (lbs/MMBtu)	0.03-0.1	0.08	-	0.1 – 0.28	0.08	-

Note: all emissions values are without post-combustion treatment.
Source: EPA, Compilation of Air Pollutant Emission Factors, AP-42, Fifth Edition, Volume I: Stationary Point and Area Sources

Boiler Emissions Control Options - NO_x

NO_x control has been the primary focus of emission control research and development in boilers. The following provides a description of the most prominent emission control approaches.

Combustion Process Emissions Control

Combustion control techniques are less costly than post-combustion control methods and are often used on industrial boilers for NO_x control. Control of combustion temperature has been the principal focus of combustion process control in boilers. Combustion control requires tradeoffs – high temperatures favor complete burn up of the fuel and low residual hydrocarbons and CO, but promote NO_x formation. Lean combustion dilutes the combustion process and reduces combustion temperatures and NO_x formation, and allows a higher compression ratio or peak firing pressures resulting in higher efficiency. However, if the mixture is too lean, misfiring and incomplete combustion occurs, increasing CO and VOC emissions.

Flue Gas Recirculation (FGR)

FGR is the most effective technique for reducing NO_x emissions from industrial boilers with inputs below 100 MMBtu/hr. With FGR, a portion of the relatively cool boiler exhaust gases re-enter the combustion process, reducing the flame temperature and associated thermal NO_x formation. It is the most popular and effective NO_x reduction method for firetube and watertube boilers, and many applications can rely solely on FGR to meet environmental standards.

External FGR employs a fan to recirculate the flue gases into the flame, with external piping carrying the gases from the stack to the burner. A valve responding to boiler input controls the recirculation rate. Induced FGR relies on the combustion air fan for flue gas recirculation. A portion of the gases travel via ductwork or internally to the air fan, where they are premixed with combustion air and introduced into the flame through the burner. Induced FGR in newer designs uses an integral design that is relatively uncomplicated and reliable. The physical limit to NOx reduction via FGR is 80% in natural gas-fired boilers and 25% for standard fuel oils.

Low Excess Air Firing (LAE)

Excess air ensures complete combustion. However, excess air levels greater than 45% can result in increased NO_x formation, because the excess nitrogen and oxygen in the combustion air entering the flame combine to form thermal NO_x. Firing with low excess air means limiting the amount of excess air that enters the combustion process, thus limiting the amount of extra nitrogen and oxygen entering the flame. Burner design modification accomplishes this and optimization uses oxygen trim controls. LAE typically results in overall NOx reductions of 5 to 10% when firing with natural gas, and is suitable for most boilers.

Low Nitrogen Fuel Oil

NO_x formed by fuel-bound nitrogen can account for 20 to 50% of total NO_x levels in oil-fired boiler emissions. The use of low nitrogen fuels in boilers firing distillate oils is one method of reducing NO_x emissions. Such fuels can contain up to 20 times less fuel-bound nitrogen than standard No. 2 oil. NO_x reductions of up to 70% over NO_x emissions from standard No. 2 oils have been achieved in firetube boilers utilizing flue gas recirculation.

Burner Modifications
Modifying the design of standard burners to create a larger flame achieves lower flame temperatures and results in lower thermal NO_x formation. While most boiler types and sizes can accommodate burner modifications, it is most effective for boilers firing natural gas and distillate fuel oils, with little effectiveness in heavy oil-fired boilers. Also, burner modifications must be complemented with other NO_x reduction methods, such as flue gas recirculation, to comply with the more stringent environmental regulations. Achieving low NO_x levels (30 ppm) through burner modification alone can adversely impact boiler operating parameters such as turndown, capacity, CO levels, and efficiency.

Water/Steam Injection
Injecting water or steam into the flame reduces flame temperature, lowering thermal NO_x formation and overall NO_x emissions. However, under normal operating conditions, water/steam injection can lower boiler efficiency by 3 to 10%. Also, there is a practical limit to the amount that can be injected without causing condensation-related problems. This method is often employed in conjunction with other NO_x control techniques such as burner modifications or flue gas recirculation. When used with natural gas-fired boilers, water/steam injection can result in NO_x reduction of up to 80%, with lower reductions achievable in oil-fired boilers.

Post-Combustion Emissions Control

There are several types of exhaust gas treatment processes that are applicable to industrial boilers.

Selective Non-Catalytic Reduction (SNCR)
In boiler SNCR, a NO_x reducing agent such as ammonia or urea is injected into the boiler exhaust gases at a temperature in the 1,400 to 1,600°F range. The agent breaks down the NO_x in the exhaust gases into water and atmospheric nitrogen (N_2). While SNCR can reduce boiler NO_x emissions by up to 70%, it is difficult to apply to industrial boilers that modulate or cycle frequently because to perform properly, the agent must be introduced at a specific flue gas temperature. The location of the exhaust gases at the necessary temperature is constantly changing in a cycling boiler.

Selective Catalytic Reduction (SCR)

This technology involves the injection of the reducing agent into the boiler exhaust gas in the presence of a catalyst. The catalyst allows the reducing agent to operate at lower exhaust temperatures than SNCR, in the 500 to 1,200°F depending on the type of catalyst. NO_x reductions of up to 90% are achievable with SCR. The two agents used commercially are ammonia (NH_3 in anhydrous liquid form or aqueous solution) and aqueous urea. Urea decomposes in the hot exhaust gas and SCR reactor, releasing ammonia. Approximately 0.9 to 1.0 moles of ammonia is required per mole of NO_x at the SCR reactor inlet in order to achieve an 80 to 90% NO_x reduction.

SCR is however costly to use and can only occasionally be justified on boilers with inputs of less than 100 MMBtu/hr. SCR requires on-site storage of ammonia, a hazardous chemical. In addition, ammonia can "slip" through the process unreacted, contributing to environmental health concerns.

Boiler Emissions Control Options - SO_x

The traditional method for controlling SO_x emissions is dispersion via a tall stack to limit ground level emissions. The more stringent SO_x emissions requirements in force today demand the use of reduction methods as well. These include use of low sulfur fuel, desulfurizing fuel, and flue gas desulfurization (FGD). Desulfurization of fuel primarily applies to coal, and, like FGD, is principally used for utility boiler emissions control. Use of low sulfur fuels is the most cost effective SO_x control method for industrial boilers, as it does not require installation and maintenance of special equipment.

FGD systems are of two types: non-regenerable and regenerable. The most common, non-regenerable, results in a waste product that requires proper disposal. Regenerable FGD converts the waste product into a product that is saleable, such as sulfur or sulfuric acid. FGD reduces SO_x emissions by up to 95%.

Technology Characterization: Fuel Cells

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Technology Characterization – Fuel Cell Systems

Introduction and Summary

Fuel cell systems, currently in the early stages of development, are an entirely different approach to the production of electricity than traditional prime mover technologies. Fuel cells are similar to batteries in that both produce a direct current (DC) through an electrochemical process without direct combustion of a fuel source. However, whereas a battery delivers power from a finite amount of stored energy, fuel cells can operate indefinitely provided the availability of a continuous fuel source. Two electrodes (a cathode and anode) pass charged ions in an electrolyte to generate electricity and heat. A catalyst enhances the process.

Fuel cells offer the potential for clean, quiet, and efficient power generation. As with most new technologies, fuel cell systems face a number of formidable market entry issues resulting from product immaturity, over-engineered system complexities, and unproven product durability and reliability. These translate into high capital cost, lack of support infrastructure, and technical risk for early adopters. However, the many advantages of fuel cells suggest that they could well become the prime mover of choice for certain applications and products in the future.

The inventor of fuel cell technology is Sir William Grove, who demonstrated a hydrogen fuel cell in London in the 1830s. Grove's technology remained without a practical application for 100 years. Fuel cells returned to the laboratory in the 1950s when the United States space program required the development of new power systems. Today, the topic of fuel cells encompasses a broad range of different technologies, technical issues, and market dynamics that make for a complex but potentially promising outlook. Significant amounts of public and private investment are being applied to the development of fuel cell products for both stationary and transportation applications.

There are five types of fuel cells under development. These are: 1) phosphoric acid (PAFC), 2) proton exchange membrane (PEMFC), 3) molten carbonate (MCFC), 4) solid oxide (SOFC), and 5) alkaline (AFC). The electrolyte and operating temperatures distinguish each type. Operating temperatures range from near ambient to 1,800°F, and electrical generating efficiencies range from 30 to over 50% HHV.[1] As a result, they can have different performance characteristics, advantages and limitations, and therefore will be suited to distributed generation applications in a variety of approaches.

The different fuel cell types share certain important characteristics. First, fuel cells are not Carnot cycle (thermal energy based) engines. Instead, they use an electrochemical or battery-like process to convert the chemical energy of hydrogen into water and electricity and can achieve high electrical efficiencies. The second shared feature is that they use hydrogen as their

[1]Most of the efficiencies quoted in this report are based on higher heating value (HHV), which includes the heat of condensation of the water vapor in the products. In engineering and scientific literature the lower heating value (LHV – which does not include the heat of condensation of the water vapor in the products) is often used. The HHV is greater than the LHV by approximately 10% with natural gas as the fuel (i.e., 50% LHV versus 45% HHV).

fuel, which is typically derived from a hydrocarbon fuel such as natural gas. Third, each fuel cell system is composed of three primary subsystems: 1) the fuel cell stack that generates direct current electricity; 2) the fuel processor that converts the natural gas into a hydrogen-rich feed stream; and 3) the power conditioner that processes the electric energy into alternating current or regulated direct current. Finally, all types of fuel cells have low emissions profiles. This is because the only combustion processes are the reforming of natural gas or other fuels to produce hydrogen and the burning of a low energy hydrogen exhaust stream that is used to provide heat to the fuel processor.

Today, there are only two commercially available fuel cells, a 200 kW PAFC unit[2] and a 250 kW MCFC unit.[3] With over 200 units sold, the PAFC fleet has achieved over 5 million operating hours in a variety of distributed generation applications. These range from a New York City police station to a major postal facility in Alaska and a credit card processing system in Nebraska. Located in over 15 countries, this initial commercial fuel cell product has successfully introduced the capabilities and features of fuel cells into the distributed generation marketplace. The MCFC fleet is more limited, with half a dozen commercial units, but several more in the development stages. While nearly two dozen companies are currently field testing a variety of alternative fuel cell systems for market entry, the availability of a wide array of off-the-shelf, fully warranted fuel cell systems designed for broad customer classes is still several years away.

Applications

Fuel cell systems are envisioned to serve a variety of distributed generation applications and markets. Since all fuel cells are in an early stage of development, there is limited experience to validate those applications considered most competitive for fuel cells. This early stage of development and commercial use causes fuel cells to be high in capital cost and to have a higher project risk due to unproven durability and reliability. These two characteristics will force introductory fuel cell products into specific markets and applications that are most tolerant of risk due to other market or operational drivers.

In DG markets, the primary characteristic driving early market acceptance is the ability of fuel cell systems to provide reliable premium power. The primary interest drivers have been their ability to achieve high efficiencies over a broad load profile and low emission signatures without additional controls.

Potential DG applications for fuel cell systems include combined heat and power (CHP), premium power, remote power, grid support, and a variety of specialty applications. **Figure 1** illustrates two actual sites with fuel cell systems functioning in DG applications.

[2] Sold by UTC Fuel Cells as the PC25, www.utcfuelcells.com.
[3] Sold by FuelCell Energy as the DFC® 300, www.fce.com.

Figure 1. FuelCell Energy Molten Carbonate Fuel Cells in Distributed Generation

Applications

Source: *www.fuelcells.org/pics.*

Combined Heat and Power

Due to the high installed cost of fuel cell systems, the most prevalent DG application envisioned by product development leaders is CHP. CHP applications are on-site power generation in combination with the recovery and use of by-product heat. Continuous baseload operation and the effective use of the thermal energy contained in the exhaust gas and cooling subsystems enhance the economics of on-site generation applications.

Heat is generally recovered in the form of hot water or low-pressure steam (<30 psig), but the quality of heat is dependent on the type of fuel cell and its operating temperature. The one exception to this is the PEM fuel cell, which operates at temperatures below 200°F, and therefore has only low quality heat. Generally, the heat recovered from fuel cell CHP systems is appropriate for low temperature process needs, space heating, and potable water heating. In the case of SOFC and MCFC technologies, medium pressure steam (up to about 150 psig) can be generated from the fuel cell's high temperature exhaust gas, but the primary use of this hot exhaust gas is in recuperative heat exchange with the inlet process gases.

The simplest thermal load to supply is hot water. Primary applications for CHP in the commercial/institutional sectors are those building types with relatively high and coincident electric and hot water/space heating demand such as colleges and universities, hospitals and nursing homes, and lodging. Technology developments in heat activated cooling/refrigeration and thermally regenerated desiccants will enhance fuel cell CHP applications by increasing the thermal energy loads in certain building types. Use of these advanced technologies in applications such as restaurants, supermarkets, and refrigerated warehouses provides a base-thermal load that opens these applications to CHP.

Premium Power

Consumers who require higher levels of reliability or power quality, and are willing to pay for it, often find some form of DG to be advantageous. These consumers are typically less concerned about the initial prices of power generating equipment than other types of consumers. Premium power systems generally supply base load demand. As a result, and in contrast to back-up generators, emissions and efficiency become more significant decision criteria.

Fuel cell systems offer a number of intrinsic features that make them suitable for the premium power market. These market-driving features include low emissions/vibration/noise, high availability, good power quality, and compatibility with zoning restrictions. As emissions become more relevant to a business's bottom line in the form of zoning issues and emissions credits, the fuel cell becomes a more appealing type of DG.

Some types of fuel cell systems have already demonstrated high availability and reliability. As fuel cells further mature in the market, they are expected to achieve the high reliability associated with fewer moving parts.

While the fuel cell requires significant power conditioning equipment in the form of direct current to alternating current conversion, power from fuel cell systems is clean, exhibiting none of the signal disturbances observed from grid sources.

Finally, zoning issues for fuel cell systems are quite possibly the least problematic of all DG systems. Fuel cell systems can be designed for both indoor and outdoor installation, and in close proximity to sensitive environments, people, or animals.

Remote Power

In locations where power from the local grid is unavailable or extremely expensive to install, DG is a competitive option. As with premium power, remote power applications are generally base load operations. Consequently, emissions and efficiency become more significant criteria in much of the remote power DG market. Coupled with their other potential advantages, fuel cell systems can provide competitive energy into certain segments of the remote power DG market. Where fuel delivery is problematic, the high efficiency of fuel cell systems can also be a significant advantage.

Grid Support

One of the first applications that drew the attention of electric utilities to fuel cell technologies was grid support. Numerous examples of utility-owned and operated distributed generating systems exist in the U.S. and abroad. The primary application in the U.S. has been the use of relatively large diesel or natural gas engines for peaking or intermediate load service at municipal utilities and electric cooperatives. These units provide incremental peaking capacity and grid support for utilities at substations. Such installations can defer the need for T&D system expansion, can provide temporary peaking capacity within constrained areas, or be used for system power factor correction and voltage support, thereby reducing costs for both customers and the utility system. The unique feature of fuel cell systems is the use of power conditioning inverters to transform direct current electricity into alternating current. These power conditioners can be operated almost independent of the fuel cell to correct power factors and harmonic characteristics in support of the grid.

Standby Power

Fire and safety codes require standby power systems for hospitals, elevator loads, and water pumping. Standby is an economic choice for customers with high outage costs such as those in the telecommunications, retail, gaming, banking, and certain process industries. The standby engine-driven generator set is typically the simplest distributed generation system, providing power only when the primary source is out of service or falters in its voltage or frequency. This application requires low capital cost, minimal installation costs, rapid black start capability, onsite fuel storage, and grid-isolated operation. In standby power applications, efficiency, emissions, and variable maintenance costs are usually not major factors in technology selection. Based on this definition of standby power, fuel cells do not appear to have much application. Fuel cell systems are characteristically high in capital cost and do not have rapid black start capability.

Peak Shaving

In certain areas of the country, customers and utilities are using on-site power generation to reduce the need for costly peak-load power. Peak shaving is also applicable to customers with poor load factor and/or high demand charges. Typically, peak shaving does not involve heat recovery, but heat recovery may be warranted where the peak period is more than 2,000 hours/year. Since low equipment cost and high reliability are the primary requirements, equipment such as reciprocating engines are ideal for many peak-shaving applications. Emissions may be an issue if operating hours are high. Combining peak shaving and another function, such as standby power, enhances the economics. High capital cost and relatively long start-up times (particularly for MCFC and SOFC) will most likely prevent the widespread use of fuel cells in peak shaving applications.

Technology Description

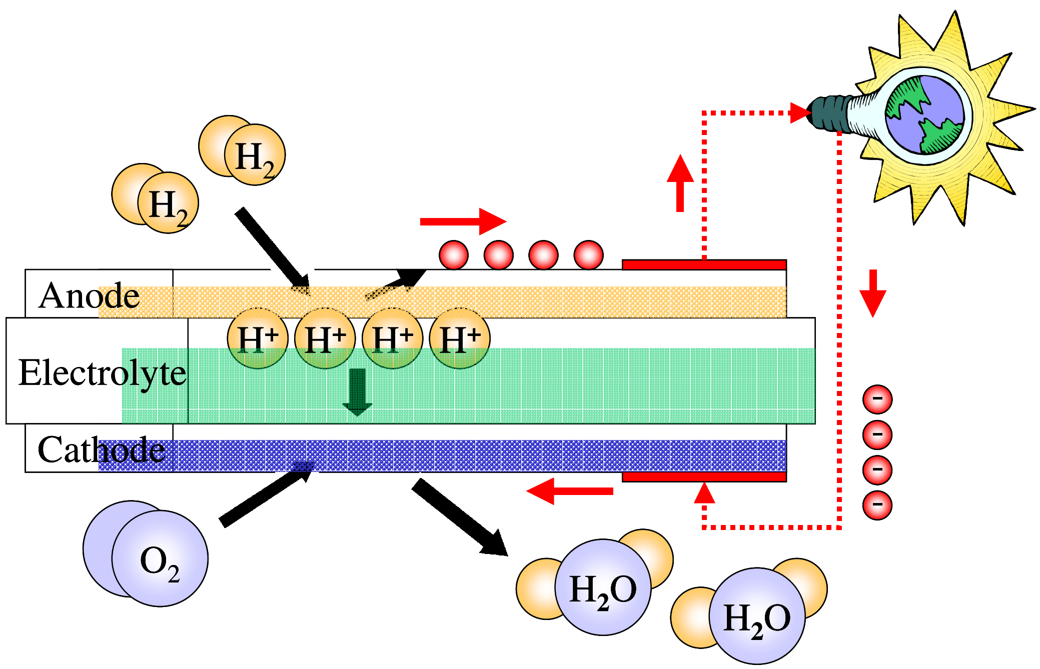
Fuel cells produce direct current electricity through an electrochemical process, much like a standard battery. Unlike a standard battery, a fuel supply continuously replenishes the fuel cell. The reactants, most typically hydrogen and oxygen gas, are fed into the fuel cell reactor, and power is generated as long as these reactants are supplied. The hydrogen (H_2) is typically generated from a hydrocarbon fuel such as natural gas or LPG, and the oxygen (O_2) is from ambient air.

Basic Processes and Components

Fuel cell systems designed for DG applications are primarily natural gas or LPG fueled systems. Each fuel cell system consists of three primary subsystems: 1) the fuel cell stack that generates direct current electricity; 2) the fuel processor that converts the natural gas into a hydrogen rich feed stream; and 3) the power conditioner that processes the electric energy into alternating current or regulated direct current.

Figure 2 illustrates the electrochemical process in a typical single cell, acid-type fuel cell. A fuel cell consists of a cathode (positively charged electrode), an anode (negatively charged electrode), an electrolyte and an external load. The anode provides an interface between the fuel and the electrolyte, catalyzes the fuel reaction, and provides a path through which free electrons conduct to the load via the external circuit. The cathode provides an interface between the oxygen and the electrolyte, catalyzes the oxygen reaction, and provides a path through which free electrons conduct from the load to the oxygen electrode via the external circuit. The electrolyte, an ionic conductive (non-electrically conductive) medium, acts as the separator between hydrogen and oxygen to prevent mixing and the resultant direct combustion. It completes the electrical circuit of transporting ions between the electrodes.

Figure 2. Fuel Cell Electrochemical Process



Source: Energy Nexus Group.

The hydrogen and oxygen are fed to the anode and cathode, respectively. The hydrogen and oxygen gases do not directly mix and combustion does not occur. Instead, the hydrogen oxidizes one molecule at a time, in the presence of a catalyst. Because the reaction is controlled at the molecular level, there is no opportunity for the formation of NO_x and other pollutants.

At the anode the hydrogen gas is electrochemically dissociated (in the presence of a catalyst) into hydrogen ions (H^+) and free electrons (e^-).

Anode Reaction: $2H_2 \rightarrow 4H^+ + 4e^-$

The electrons flow out of the anode through an external electrical circuit. The hydrogen ions flow into the electrolyte layer and eventually to the cathode, driven by both concentration and potential forces. At the cathode the oxygen gas is electrochemically combined (in the presence of a catalyst) with the hydrogen ions and free electrons to generate water.

Cathode Reaction: $O_2 + 4H^+ + 4e^- \rightarrow 2H_2O$

The overall reaction in a fuel cell is as follows:

Net Fuel Cell Reaction: $2H_2 + O_2 \rightarrow 2H_2O$ (vapor) + Energy

The amount of energy released is equal to the difference between the Gibbs free energy of the product and the Gibbs free energy of the reactants.

When generating power, electrons flow through the external circuit, ions flow through the electrolyte layer and chemicals flow into and out of the electrodes. Each process has natural resistances, and overcoming these reduces the operational cell voltage below the theoretical potential. There are also irreversibilities[4] that affect actual open circuit potentials. Therefore, some of the chemical potential energy converts into heat. The electrical power generated by the fuel cell is the product of the current measured in amps and the operational voltage. Based on the application and economics, a typical operating fuel cell will have an operating voltage of between 0.55 volts and 0.80 volts. The ratio of the operating voltage and the theoretical maximum of 1.48 volts represents a simplified estimate of the stack electrical efficiency on a higher heating value (HHV[5]) basis.

As explained, resistance heat is also generated along with the power. Since the electric power is the product of the operating voltage and the current, the quantity of heat that must be removed from the fuel cell is the product of the current and the difference between the theoretical potential and the operating voltage. In most cases, the water produced by the fuel cell reactions exits the fuel cell as vapor, and therefore, the 1.23-volt LHV theoretical potential is used to estimate sensible heat generated by the fuel cell electrochemical process.

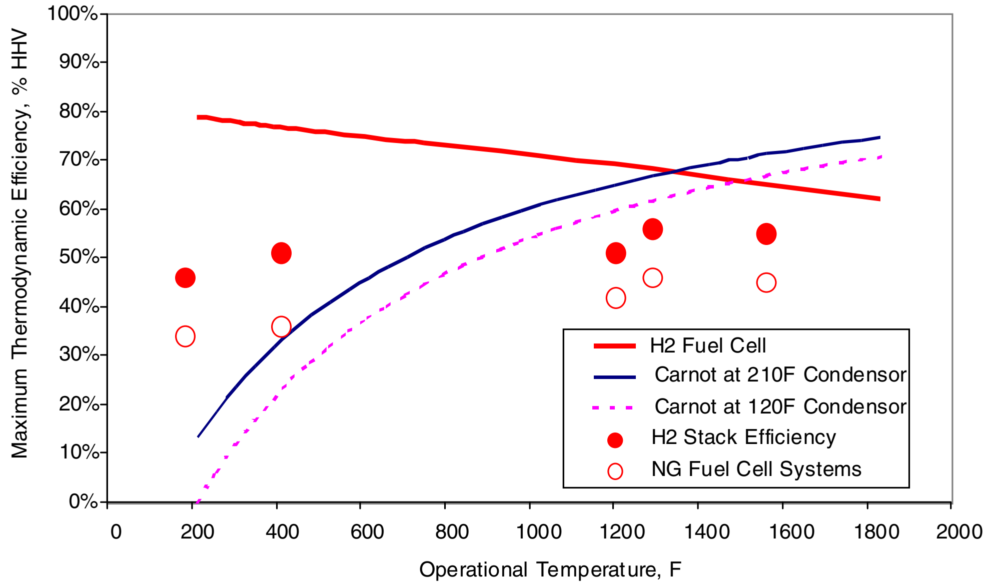
The overall electrical efficiency of the cell is the ratio of the power generated and the heating value of the hydrogen consumed. The maximum thermodynamic efficiency of a hydrogen fuel cell is the ratio of the Gibbs free energy and the heating value of the hydrogen. The Gibbs free energy decreases with increasing temperatures, because the product water produced at the elevated temperature of the fuel cell includes the sensible heat of that temperature, and this energy cannot be converted into electricity without the addition of a thermal energy conversion cycle (such as a steam turbine). Therefore, the maximum efficiency of a pure fuel cell system decreases with increasing temperature. **Figure 3** illustrates this characteristic in comparison to

[4] Irreversibilities are changes in the potential energy of the chemical that are not reversible through the electrochemical process. Typically, some of the potential energy is converted into heat even at open circuit conditions when current is not flowing. A simple example is the resistance to ionic flow through the electrolyte while the fuel cell is operating. This potential energy “loss” is really a conversion to heat energy, which cannot be reconverted into chemical energy directly within the fuel cell.

[5] Most of the efficiencies quoted in this report are based on higher heating value (HHV), which includes the heat of condensation of the water vapor in the products.

the Carnot cycle efficiency limits through a condenser at 50 and 100°C.[6] This characteristic has led system developers to investigate hybrid fuel cell-turbine combined cycle systems to achieve system electrical efficiencies in excess of 70% HHV.

Figure 3. Effect of Operating Temperature on Fuel Cell Efficiency



Source: Siemens/Westinghouse Electric Corp.

Fuel Cell Stacks

Practical fuel cell systems require voltages higher than 0.55 to 0.80. Combining several cells in electrical series into a fuel cell stack achieves this. Typically, there are several hundred cells in a single cell stack. Increasing the active area of individual cells manages current flow. Typically, cell area can range from 100 cm^2 to over 1 m^2 depending on the type of fuel cell and application power requirements.

Fuel Processors

In distributed generation applications, the most viable fuel cell technologies use natural gas as the system's fuel source. To operate on natural gas or other fuels, fuel cells require a fuel processor or reformer, a device that converts the fuel into the hydrogen-rich gas stream. While adding fuel flexibility to the system, the reformer also adds significant cost and complexity. There are three primary types of reformers: steam reformers, autothermal reformers, and partial oxidation reformers. The fundamental differences are the source of oxygen used to combine with the carbon within the fuel to release the hydrogen gases and the thermal balance of the chemical process. Steam reformers use steam, while partial oxidation units use oxygen gas, and autothermal reformers use both steam and oxygen

[6] Larminie, James and Andrew Dicks, Fuel Cell Systems Explained. John Wiley & Sons, Ltd., West Sussex, England, 2000.

Steam reforming is extremely endothermic and requires a substantial amount of heat input. Autothermal reformers typically operate at or near the thermal neutral point, and therefore, do not generate or consume thermal energy. Partial oxidation units combust a portion of the fuel (i.e. partially oxidize it), releasing heat in the process. When integrated into a fuel cell system that allows the use of anode-off gas, a typical natural gas reformer can achieve conversion efficiencies in the 75 to 90% LHV range, with 83 to 85% being an expected level of performance. These efficiencies are defined as the LHV of hydrogen generated divided by the LHV of the natural gas consumed by the reformer.

Some fuel cells can function as *internally steam reforming* fuel cells. Since the reformer is an endothermic catalytic converter and the fuel cell is an exothermic catalytic oxidizer, the two combine into one with mutual thermal benefits. More complex than a pure hydrogen fuel cell, these types of fuel cells are more difficult to design and operate. While combining two catalytic processes is difficult to arrange and control, these internally reforming fuel cells are expected to account for a significant market share as fuel cell based DG becomes more common.

Power Conditioning Subsystem

The fuel cell generates direct current electricity, which requires conditioning before serving a DG application. Depending on the cell area and number of cells, this direct current electricity is approximately 200 to 400 volts per stack. If the system is large enough, stacks can operate in series to double or triple individual stack voltages. Since the voltage of each individual cell decreases with increasing load or power, the output is considered an unregulated voltage source. The power conditioning subsystem boosts the output voltage to provide a regulated higher voltage input source to an electronic inverter. The inverter then uses a pulse width modulation technique at high frequencies to generate simulated alternating current output. The inverter controls the frequency of the output, which can be adjusted to enhance power factor characteristics. Because the inverter generates alternating current within itself, the output power is generally clean and reliable. This characteristic is important to sensitive electronic equipment in premium power applications. The efficiency of the power conditioning process is typically 92 to 96%, and is dependent on system capacity and input voltage-current characteristic.

Types of Fuel Cells

There are five basic types of fuel cell under consideration for DG applications. The fuel cell's electrolyte or ion conduction material defines the basic type. Two of these fuel cell types, polymer electrolyte membrane (PEM) and phosphoric acid fuel cell (PAFC), have acidic electrolytes and rely on the transport of H^+ ions. Two others, alkaline fuel cell (AFC) and carbonate fuel cell (MCFC), have basic electrolytes that rely on the transport of OH^- and CO_3^{2-} ions, respectively. The fifth type, solid oxide fuel cell (SOFC), is based on a solid-state ceramic electrolyte in which oxygen ions (O^{2-}) are the conductive transport ion.

Each fuel cell type operates at optimum temperature, which is a balance between the ionic conductivity and component stability. These temperatures differ significantly among the five basic types, ranging from near ambient to as high as 1800°F. The proton conducting fuel cell type generates water at the cathode and the anion conducting fuel cell type generates water at the anode. **Table 1** below presents fundamental characteristics for each fuel cell type.

Table 1. Characteristics of Major Fuel Cell Types

	PEMFC	**AFC**	**PAFC**	**MCFC**	**SOFC**
Type of Electrolyte	H^+ ions (with anions bound in polymer membrane)	OH^- ions (typically aqueous KOH solution)	H^+ ions (H_3PO_4 solutions)	CO_3^{2-} ions (typically, molten $LiKaCO_3$ eutectics)	O^{2-} ions (Stabilized ceramic matrix with free oxide ions)
Typical construction	Plastic, metal, or carbon	Plastic, metal	Carbon, porous ceramics	High temp metals, porous ceramic	Ceramic, high temp metals
Internal reforming	No	No	No	Yes, Good Temp Match	Yes, Good Temp Match
Oxidant	Air to O_2	Purified Air to O_2	Air to Enriched Air	Air	Air
Operational Temperature	150- 180°F (65-85°C)	190-500°F (90-260°C)	370-410°F (190-210°C)	1,200-1,300°F (650-700°C)	1,350-1,850°F (750-1,000°C)
DG System Level Efficiency, % HHV	25 to 35%	32 to 40%	35 to 45%	40 to 50%	45 to 55%
Primary Contaminate Sensitivities	CO, Sulfur, and NH3	CO, CO_2, and Sulfur	CO < 1%, Sulfur	Sulfur	Sulfur

Source: Energy Nexus Group

PEMFC (Proton Exchange Membrane Fuel Cell or Polymer Electrolyte Membrane)

NASA developed this type of fuel cell in the 1960s for the first manned spacecraft. The PEMFC uses a solid polymer electrolyte and operates at low temperatures (about 200°F). Over the past ten years, the PEMFC has received significant media coverage due to the large auto industry investment in the technology. Due to their modularity and potential for simple manufacturing, reformer/PEMFC systems for residential DG applications have attracted considerable development capital. PEMFC's have high power density and can vary their output quickly to meet demand. This type of fuel cell is highly sensitive to CO poisoning.

AFC (Alkaline Fuel Cell)

F.T. Bacon in Cambridge, England first demonstrated AFC as a viable power unit during the 1940s and 1950s. NASA later developed and used this fuel cell on the Apollo spacecraft and on the space shuttles. AFC technology uses alkaline potassium hydroxide as the electrolyte. The primary advantages of AFC technology are improved performance (electrical efficiencies above 60% HHV), use of non-precious metal electrodes, and the fact that no unusual materials are needed. The primary disadvantage is the tendency to absorb carbon dioxide, converting the alkaline electrolyte to an aqueous carbonate electrolyte that is less conductive. The attractiveness of AFC has declined substantially with the interest and improvements in PEMFC technology.

PAFC (Phosphoric Acid Fuel Cell)

PAFC uses phosphoric acid as the electrolyte and is generally considered the most established fuel cell technology. The first PAFC DG system was designed and demonstrated in the early 1970s. PAFCs are capable of fuel-to-electricity efficiencies of 36% HHV or greater. A 200 kW PAFC has been commercially available since the early 1990s. Over 200 of these commercial units were manufactured, delivered, and are operating in the U.S., Europe, and Japan. The current 200 kW product is reported to have a stack lifetime of over 40,000 hours, units with nearly eight years of operation, and commercially based reliabilities in the 90 to 95% range. The major market barrier has been the initial installed cost that has not yet fallen below the $4,500 to $5,500/kW range.

<u>MCFC (Molten Carbonate Fuel Cell)</u>
The MCFC uses an alkali metal carbonate (Li, Na, K) as the electrolyte and has a developmental history that dates back to the early part of the twentieth century. Due to its operating temperature range of 1,100 to 1,400°F, the MCFC holds promise in both CHP and DG applications. This type of fuel cell can be internally reformed, can operate at high efficiencies (50% HHV), and is relatively tolerant of fuel impurities. Government/industry R&D programs during the 1980s and 1990s resulted in several individual pre-prototype system demonstrations. FuelCell Energy is presently the only company with a commercial molten carbonate fuel cell. The primary technical issue with MCFC technologies is the degradation of cell components due to the corrosive nature of the electrolyte/operating temperature combination.

<u>SOFC (Solid Oxide Fuel Cell)</u>
The SOFC uses solid, nonporous metals oxide electrolytes and is generally considered less mature in its development than the MCFC and PAFC technologies. Several SOFC units up to 100 kW in size and based on a concentric tubular design have been built and tested.[7] In addition, there are many companies developing planar SOFC designs, which offer higher power densities and lower costs than the tubular design, but these have yet to achieve the reliability of the tubular design. Despite relative immaturity, the SOFC has several advantages (high efficiency, stability and reliability, and high internal temperatures) that have attracted development support. The SOFC has projected service electric efficiencies of 45 to 60% and higher, for larger hybrid, combined cycle plants. Efficiencies for smaller SOFC DG units are expected to be in the 50% range.

Stability and reliability of the SOFC are due to an all-solid-state ceramic construction. Test units have operated in excess of 10 years with acceptable performance. The high internal temperatures of the SOFC are both an asset and a liability. As an asset, high temperatures make internal reforming possible. As a liability, these high temperatures add to materials and mechanical design difficulties, which reduces stack life and increases cost. While SOFC research has been ongoing for 30 years, costs of these stacks are still comparatively high.

Design Characteristics

The features that have the potential to make fuel cell systems a leading prime mover for CHP and other distributed generation applications include:

[7] By Siemens/Westinghouse Electric Corp.

Size range: Fuel cell systems are constructed from individual cells that generate 100 W to 2 kW per cell. This allows systems to have extreme flexibility in capacity. Systems under development for DG application range in sizes from 5 kW to 2 MW. Multiple systems can operate in parallel at a single site to provide incremental capacity.

Thermal output: Fuel cells can achieve overall efficiencies in the 65 to 85% range. Waste heat can be used primarily for domestic hot water applications and space heating.

Availability: The commercially available 200 kW PC25 system fleet (200-plus units) has demonstrated greater than 90% availability during over four million operating hours. As fuel cell systems mature, their reliability should improve.

Part-load operation: Fuel cell stack efficiency improves at lower loads, which results in a system electric efficiency that is relatively steady down to one-third to one-quarter of rated capacity. This provides systems with excellent load following characteristics.

Cycling: While part-load efficiencies of fuel cells are generally high, MCFC and SOFC fuel cells require long heat-up and cool-down periods, restricting their ability to operate in many cyclic applications.

High quality power: Electrical output is computer grade power, meeting critical power requirements without interruption. This minimizes lost productivity, lost revenues, product loss, or opportunity cost.

Reliability and life: Since only auxiliary components have moving parts, the reliability of fuel cells is expected to be high. A few of the initial PC25 systems have achieved operational lives of 70,000 hours.

Emissions: The only combustion within a fuel cell system is the low energy content hydrogen stream exhausted from the stack. This stream is combusted within the reformer and can achieve emissions signatures of < 2 ppmv CO, <1 ppmv NO_x and negligible SO_x (on 15% O_2, dry basis).

Efficiency: Different types of fuel cells have varied efficiencies. Depending on the type and design of fuel cells, electric efficiency ranges from 30% to over 50% HHV.

Quiet Operation: Conversational level (60dBA @ 30 ft.), acceptable for indoor installation.

Siting and Size:	Indoor or outdoor installation.
Fuel Use:	The primary fuel source for the fuel cell is hydrogen, which can be obtained from natural gas, coal gas, methanol, and other fuels containing hydrocarbons.

Performance Characteristics

Fuel cell performance is a function of the type of fuel cell and its capacity. Since the fuel cell system is a series of chemical, electrochemical, and electronic subsystems, the optimization of electric efficiency and performance characteristics can be a challenging engineering task. The electric efficiency calculation example provided in the next section illustrates this.

Table 2 summarizes performance characteristics for representative commercially available and developmental natural gas fuel cell CHP systems over the 10 kW to 2 MW size range. This size range covers the majority of the market applications currently envisioned for fuel cell CHP and represents the most likely units to be commercially introduced within the next five years. Of the systems included in **Table 2**, the only commercially available at this time is the PAFC product, first introduced in 1992. The other systems are in various phases of prototype demonstration. Cost and performance estimates for these systems are based on initial market entry targets. The capital cost estimate for the PAFC system represents published cost from the manufacturer for lots of three or more units. Since the other systems are just emerging from their demonstration phases, pricing and costing information are subjective and estimates should be considered within the plus or minus thirty percent range.

Table 2. Fuel Cell CHP - Typical Performance Parameters

Cost and Performance Characteristics[8]	**System 1**	**System 2**	**System 3**	**System 4**	**System 5**	**System 6**
Fuel Cell Type	PAFC	PEM	PEM	MCFC	MCFC	SOFC
Nominal Electricity Capacity (kW)	200	10	200	250	2,000	100
Commercial Status 2002[9]	Com'l	Demo	Demo	Demo	Demo	Demo
Operating Temperature (°F)	400	150	150	1200	1200	1750
Package Cost (2002 $/kW) [10]	3,850	4,700	2,950	4,350	2,400	2,850
Total Installed Cost (2002 $/kW) [11]	4,500	5,500	3,600	5,000	2,800	3,500
O&M Costs ($/kW) [12]	0.029	0.033	0.023	0.043	0.033	0.023
Electric Heat Rate (Btu/kWh)[13]	9,480	11,370	9,750	7,930	7,420	7,580
Electrical Efficiency (% HHV)[14]	36%	30%	35%	43%	46%	45%
Fuel Input (MMBtu/hr)	1.90	0.10	2.00	2.00	14.80	0.80
CHP Characteristics						
Heat Avail. >160°F (MMBtu/hr)	0.37	0.00	0.00	0.22	1.89	0.10
Heat Avail. <160°F (MMBtu/hr)	0.37	0.04	0.72	0.22	1.67	0.09
Heat Output (MMBtu/hr)	0.74	0.04	0.72	0.44	3.56	0.19
Heat Output (kW equivalent)	217	13	211	128	1043	56
Total CHP Efficiency (%), HHV[15]	75%	68%	72%	65%	70%	70%
Power/Heat Ratio[16]	0.92	0.77	0.95	1.95	1.92	1.79
Net Heat Rate (Btu/kWh)[17]	4,860	6,370	5,250	5,730	5,200	5,210
Effective Electrical Eff (%), HHV	70.3%	53.6%	65.0%	59.5%	65.7%	65.6%

Source: Energy Nexus Group.

[8] Data are representative typical values for developmental systems based on available information from fuel cell system developers. Only the PAFC data are representative of a commercial product available in 2002. Developers include but are not limited to UTC Fuel Cells, Toshiba, Ballard Power, Plug Power, Fuel Cell Energy, Siemens-Westinghouse, H-Power, Hydrogenics, Honeywell, Fuji, IHI, Global Thermal, Mitsubishi Heavy Industries, and Ztek.

[9] Com'l = Commercially Available; Demo = Multiple non-commercial demonstrations completed or underway in field sites with potential customers; Lab = Characteristics observed in laboratory validation testing of complete systems; Exp = Only experimental prototypes have been tested.

[10] Packaged Cost includes estimates of typical costs for a CHP compatible system with grid interconnection functionality built into power conditioning subsystem.

[11] Total Installed Cost include estimates for packaged cost plus electrical isolation equipment, hot water CHP interconnections, site labor and preparation, construction management, engineering, contingency, and interest during construction. See Table 3.

[12] O&M costs are estimated based on service contract nominal rate, consumables, fixed costs, and sinking fund for stack replacement at end of life. See Table 4.

[13] All equipment manufacturers quote heat rates in terms of the lower heating value (LHV) of the fuel. On the other hand, the usable energy content of fuels is typically measured on a higher heating value (HHV) basis. In addition, electric utilities measure power plant heat rates in terms of HHV. For natural gas, the average heat content of natural gas is 1,030 Btu/scf on an HHV basis and 930 Btu/scf on an LHV basis – or about a 10% difference.

[14] Electrical efficiencies are net of parasitic and conversion losses.

[15] Total Efficiency = (net electric generated + net heat produced for thermal needs)/total system fuel input

[16] Power/Heat Ratio = CHP electrical power output (Btu)/ useful heat output (Btu)

[17] Effective Electrical Efficiency = (CHP electric power output)/(Total fuel into CHP system – total heat recovered/0.8). Equivalent to 3,412 Btu/kWh/Net Heat Rate and Net Heat Rate = 3412/Effective Elec Eff.

Heat rates and efficiencies shown were taken from manufacturers' specifications, industry publications, or are based on the best available data for developing technologies. Available thermal energy was calculated from estimated overall efficiency for these systems. CHP thermal recovery estimates are based on producing low quality heat for domestic hot water process or space heating needs. This feature is generally acceptable for commercial/institutional applications where it is more common to have hot water thermal loads.

The data in the table show that electrical efficiency increases as the operating temperature and size of the fuel cell increases. As electrical efficiency increases, the absolute quantity of thermal energy available to produce useful thermal energy decreases per unit of power output, and the ratio of power to heat for the CHP system generally increases. A changing ratio of power to heat impacts project economics and may affect the decisions that customers make in terms of CHP acceptance, sizing, and the desirability of selling power.

Electrical Efficiency

As with all generation technologies, the electrical efficiency is the ratio of the power generated and the heating value of the fuel consumed. Because the fuel cell system has several subsystems in series, the electrical efficiency of the DG unit is the multiple of the efficiencies of the individual section. The concept of stack electric efficiency was introduced earlier. The following calculates the electric efficiency of a fuel cell system:

ElecEff = (FPS Eff * H_2 Utilization * Stack Eff * PC Eff)*(HHV/LHV ratio of the fuel)

Where: FPS Eff	=	Fuel Processing Subsystem Efficiency, LLV
	=	(LHV of H_2 Generated/LHV of Fuel Consumed)
H_2 Utilization	=	% of H_2 actually consumed in the stack
Stack Eff	=	(Operating Voltage/Energy Potential ~1.23 volts)
PC Eff	=	AC power delivered/(DC power generated) (auxiliary loads are assumed DC loads here)

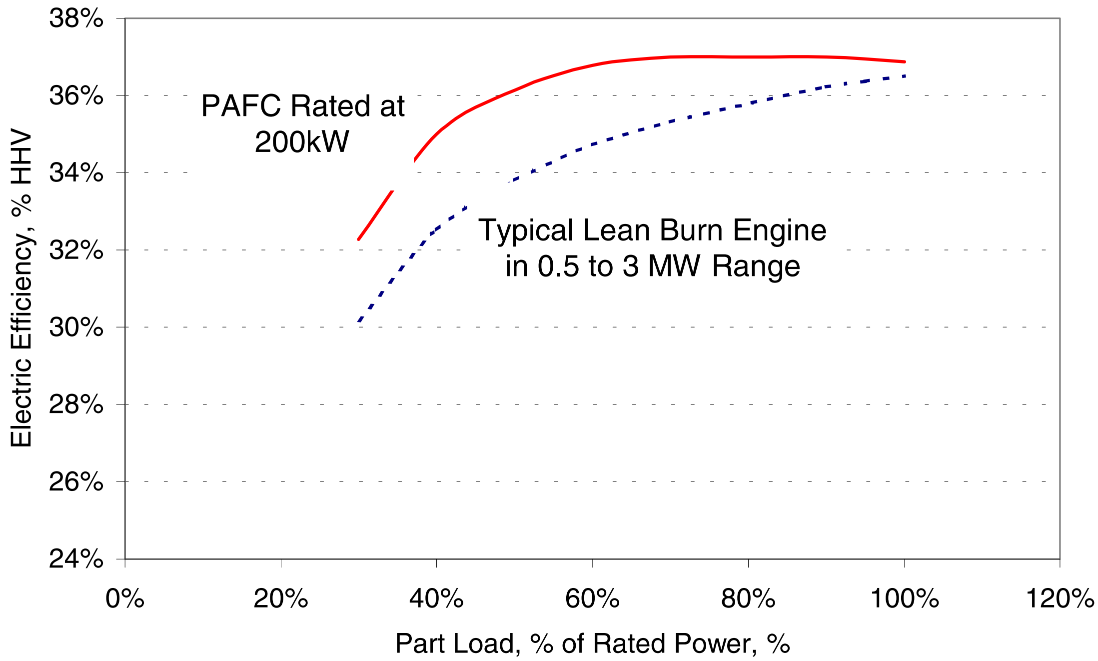
For example: PAFC = (84%FPS)*(83% util)*(0.75V/1.25V)*(95%PC)*(0.9HHV/LHV)
= 36% electric efficiency HHV

As the operating temperature range of the fuel cell system increases, the electric efficiency of the systems tends to increase. Although the maximum thermodynamic efficiency decreases as shown in **Figure 2**, improvements in reformer subsystem integration and increases in reactant activity balance out to provide the system level increase. Advanced high temperature MCFC and SOFC systems are projected to achieve simple cycle efficiencies in the range of 50 to 55% HHV, while hybrid combined fuel cell-heat engine systems are calculated to achieve efficiencies above 60% in DG applications.

Part-Load Performance

In power generation and CHP applications, fuel cell systems follow either the electric or thermal load of the applications to maximize DG energy economics. **Figure 4** shows the part-load efficiency curve for a market entry PAFC fuel cell in comparison to a typical lean burn natural gas engine. The efficiency at 50% load is within 2% of its full load efficiency characteristic. As the load decreases further, the curve becomes somewhat steeper, as inefficiencies in air blowers and the fuel processor begin to override the stack efficiency improvement.

Figure 4. Comparison of Part Load Efficiency Derate



Source: Gas Research Institute, Caterpillar, and Energy Nexus Group.

Effects of Ambient Conditions on Performance

Fuel cells are rated at ISO conditions of 77°F and 0.987 atmospheres (1 bar) pressure. Fuel cell system performance – both output and efficiency – can degrade as ambient temperature or site elevation increases. Ancillary equipment performance, primarily the air handling blowers or compressors, accounts for the degradation of performance. Performance degradations will be greater for pressurized systems operating with turbo-chargers or small air compressors as their primary air supply components.

Heat Recovery

The economics of fuel cells in on-site power generation applications depend less on effective use of the thermal energy recovered than is the case with lower efficiency prime movers, but thermal load displacements improves operating economics as in any CHP application. Generally, the stack and reformer subsystems contain 25% of the inlet fuel energy in the form of higher quality thermal energy. The exhaust gases (includes the latent heat of the product water generated in the fuel cell) contain another 25% of the recoverable energy. The most common use of this heat is to generate hot water or low-pressure steam for process use or for space heating, process needs, or domestic hot water.

Heat can generally be recovered in the form of hot water or low-pressure steam (< 30 psig), but the quality of heat is dependent on the type of fuel cell and its operating temperature. The one exception to this is the PEM fuel cell, which operates at temperatures below 100°C, and therefore has only low quality heat.

As an example, there are four primary potential sources of usable waste heat from a fuel cell system: exhaust gas including water condensation, stack cooling, anode-off gas combustion, and reformer heat. The PAFC system achieves 36% electric efficiency and 72% overall CHP efficiency, which means that it has a 36% thermal efficiency or power to heat ratio of one. Of the available heat, 25 to 45% is recovered from the stack-cooling loop that operates at approximately 400°F and can deliver low- to medium-pressure steam. The exhaust gas-cooling loop provides the balance of the heat and serves two functions. The first is condensation of product water, thus rendering the system water self-sufficient, and the second is the recovery of by-product heat. Since its primary function is water recovery, the balance of the heat available from the PAFC fuel cell is recoverable with 120°F return and 300°F supply temperatures. This tends to limit the application of this heat to domestic hot water applications. Maximum system efficiency occurs when all of the available anode-off gas heat and internal reformer heat is used internally.

In the case of SOFC and MCFC fuel cells, medium-pressure steam (up to about 150 psig) can be generated from the fuel cell's high temperature exhaust gas, but the primary use of these hot exhaust gas is in recuperative heat exchange with the inlet process gases. Like engine and turbine systems, the fuel cell exhaust gas can be used directly for process drying.

Performance and Efficiency Enhancements

Air is fed to the cathode side of the fuel cell stack to provide the oxygen needed for the power generation process. Typically, 50 to 100% more air passes through the cathode than the fuel cell reactions require. The fuel cell operated at near-ambient pressure, or at elevated pressures to enhance stack performance. Increasing the pressure, and therefore the partial pressure of the reactants, increases stack performance by reducing the electrode over potentials associated with moving the reactants into the electrodes where the catalytic reaction occurs. It also improves the performance of the catalyst. Assuming optimistic compressor characteristics, these improvements appear to optimize at approximately three atmospheres pressure.[18] More realistic assumptions often result in optimizations at ambient pressure where air movement requires the least energy. Because of these characteristics, developers focus on both pressurized and ambient pressure systems.

Capital Cost

This section provides estimates for the installed cost of fuel cell systems designed for DG applications. Only CHP configurations are presented because the majority of developments appear to integrate heat recovery capability primarily to support product water condensation and self-sufficiency. Capital costs (equipment and installation) are estimated in **Table 3** for the six typical fuel cell systems presented in **Table 1** in the 10 kW to 2 MW size range. Estimates are

[18] Ibid., p. 90.

"typical" budgetary price levels. Installed costs can vary significantly depending on the scope of the plant equipment, geographical area, competitive market conditions, special site requirements, prevailing labor rates, and whether the system is a new or retrofit application.

Based on commercially available information and internal analysis, each of five major component groups with approximately twenty major components was used to define the individual fuel cell systems and to develop total package cost estimates. Cost and pricing information was estimated in constant 2002 dollars and totaled across major component groups to achieve the estimated packaged cost of each system. This process allowed for uniform estimates for similar components of similar capacity such as the power electronics, and adjustments for components such as the cell stack and reformer subsystems due to the technology differences and product requirements

Following the above approach, each fuel cell system was broken down into the following five major component groups or subsystems:

- Stack subsystem – consisting of the fuel cell stacks, feed gas manifolds, and power take-offs.
- Fuel processing subsystem – consisting of fuel management controls, reformer, steam generators, shift reactors, sulfur absorbent beds, and ancillary components.
- Power and electronic subsystem – consisting of a solid-state boost regulator, DC to AC inverters, grid interconnect switching, load management and distribution hardware, and inverter controller and overall supervisory controller.
- Thermal management subsystem – consisting of the stack cooling system, heat recovery and condensing heat exchangers.
- Ancillary subsystems – consisting of the process air supply blowers, water treatment system, safety controls and monitoring, cabinet ventilation fans, and other miscellaneous components.

From a cost and complexity standpoint, each individual system and system developer has a different perspective on the details of these subsystems. The stack subsystem represents from 25 to 40% of equipment cost, the fuel processing subsystem from 25 to 30%, the power and electronics subsystem from 10 to 20%, the thermal management subsystem from 10 to 20%, and ancillary subsystems from 5 to 15%. One of the major issues with fuel cell systems is process complexities and the cost of equipment to maintain expanded features and characteristics.

The cost of the basic fuel cell package plus the costs for added systems needed for the particular application comprise the total equipment cost. The total plant cost consists of total equipment cost plus installation labor and materials (including site work), engineering, project management (including licensing, insurance, commissioning and startup), and financial carrying costs during construction. The installation costs of fuel cell systems are relatively consistent with engine-based equipment. The range of $400 to $800/kW used in **Table 2** reflects this similarity but include slight increases due to issues that will arise because of the uniqueness of the equipment. No additional costs were applied for emission controls technologies or permitting delays.

Table 3. Estimated Capital Cost for Typical Fuel Cell Systems in Grid Interconnected CHP Applications ($/kW)*

Installed Cost Components	System 1	System 2	System 3	System 4	System 5	System 6
Nominal Capacity (kW)	200	10	200	250	2000	100
Fuel Cell Type	PAFC	PEM	PEM	MCFC	MCFC	SOFC
Equipment Costs (2002 $/kW)						
Packaged Cost	3,850	4,700	2,950	4,350	2,400	2,850
Grid Isolation Breakers[19]	100	250	100	100	15	120
Materials and Labor[20]	272	100	272	280	230	250
Total Process Capital	4,222	5,050	3,370	4,780	2,645	2,270
Other Site Costs (2002 $/kW)[21]						
Proj. & Const. Management	124	280	124	112	80	168
Engineering and Fees	52	90	52	60	25	72
Contingencies	94	80	94	90	20	30
Interest during Construction	8	0	8	8	30	10
Total Installed Cost (2002 $/kW)	4,500	5,500	3,600	5,000	2,800	3,500

** Estimated capital costs for current technology fuel cell systems in the 2003/04 timeframe.*
Source: Energy Nexus Group.

Maintenance

Maintenance costs for fuel cell systems vary with type of fuel cell, size, and maturity of the equipment. Some of the typical costs that need to be included are:

- Maintenance labor.
- Ancillary replacement parts and material such as air and fuel filters, reformer igniter or spark plug, water treatment beds, flange gaskets, valves, electronic components, etc., and consumables such as sulfur adsorbent bed catalysts and nitrogen for shutdown purging.
- Major overhauls include shift catalyst replacement (3 to 5 years), reformer catalyst replacement (5 years), and stack replacement (4 to 8 years).

In-house personnel can perform basic maintenance, or it can be contracted out to manufacturers, distributors, or dealers under service contracts. Details of full maintenance contracts (covering all recommended service) and costing are not generally available, but are estimated at 0.7 to 2.0 cents/kWh excluding the stack replacement cost sinking fund. Maintenance for initial

[19] Only grid isolation breakers included because functionality of grid interconnection and isolation has been included into the power conditioning subsystem with package cost. For example, $100/kW was included for System 1 and System 3 at 200 kW capacity.
[20] Materials and labor estimates were based on typical engine CHP systems for units between 100 and 2,000 kW. The 10 kW system was estimated based on appliance type residential installations.
[21] Other site-related costs were estimated based on the extrapolated data from typical engine CHP systems for units between 100 and 2,000 kW. The interest during construction for the 10 kW system was set at zero because of the residential appliance-like nature of equipment.

commercial fuel cells has included remote monitoring of system performance and conditions and an allowance for predictive maintenance. Routine short interval inspections/adjustments and periodic replacement of filters (projected at intervals of 2,000 to 4,000 hours) comprise the recommended service.

Table 4 presents maintenance costs based on service contracts consisting of routine inspections and scheduled overhauls of the fuel cell system and are prorated based on comparable engine generator service contracts. Stack life and replacement costs are based on developers' estimates for initial units. Overall, maintenance costs are based on 8,000 annual operating hours expressed in terms of annual electricity generation.

Table 4. Estimated Operating and Maintenance Costs Of Typical CHP Fuel Cell Systems*

O&M Cost Analysis (2002 $)[22]	**System 1**	**System 2**	**System 3**	**System 4**	**System 5**	**System 6**
Nominal Capacity (kW)	200	10	200	250	2,000	100
Fuel Cell Type	PAFC	PEM	PEM	MCFC	MCFC	SOFC
Variable Service Contract ($/kWh)	0.0087	0.0121	0.0087	0.0072	0.0054	0.0102
Variable Consumables ($/kWh)	0.0002	0.0002	0.0002	0.0002	0.0002	0.0002
Fixed ($/kW-yr)	6.5	18.0	6.5	5.0	2.1	10.0
Fixed($/kWh @ 8,000 hrs/yr)	0.0008	0.0023	0.0008	0.0006	0.0003	0.0013
Stack Fund ($/kWh)[23]	0.0193	0.0188	0.0132	0.0350	0.0275	0.0125
Stack Life (yrs)	5	4	4	4	4	8
Recovery Factor (%)	30%	50%	35%	30%	20%	20%
Net O&M cost ($/kWh)	0.029	0.033	0.023	0.043	0.033	0.023

* *Estimated costs for current technology fuel cell systems in the 2003/04 timeframe*
Source: Energy Nexus Group.

Fuels

Fuel cell systems operate on a variety of alternative gaseous fuels since the primary fuel source is hydrogen produced from hydrocarbon fuels. These including the following:

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[22] Maintenance costs presented in **Table 4** are based on 8,000 operating hours expressed in terms of annual electricity generation. Fixed costs are based on an interpolation of engine manufacturers' estimates and applied to fuel cell system. The variable component of the O&M cost represents the inspections and minor procedures that are normally conducted by the original equipment manufacturer through a service agreement, and have been estimated based on 60% of reciprocating engine service contracts. Major overhaul procedures primarily representing stack replacements have been handled as a separate item.

[23] Stack replacement costs have been estimated = (stack original cost*(1-recovery factor))/(stack life*8000hrs/yr). Stack life was estimated based on type of fuel cell. Recovery factor was based on catalyst recovery, metal scrap value and non-repeat hardware value at end of life. All estimates are considered first cut projections and have an uncertainty of +/- one year and +/- 15%. The small PEM recovery factor was increased due to its higher non-repeat component cost.

- Liquefied petroleum gas (LPG) – propane and butane mixtures.

- Sour gas - unprocessed natural gas as it comes directly from the gas well.

- Biogas – any of the combustible gases produced from biological degradation of organic wastes, such as landfill gas, sewage digester gas, and animal waste digester gas.

- Industrial waste gases – flare gases and process off-gases from refineries, chemical plants and steel mill.

- Manufactured gases – typically low- and medium-Btu gas produced as products of gasification or pyrolysis processes.

Factors that impact the operation of a fuel cell system with alternative gaseous fuels include:

- Volumetric heating value – Since the fuel cell's fuel processing system initially reforms the fuel, the lower energy content fuels simply result in a less concentrated hydrogen-rich gas stream feeding the anode. This will cause some loss in stack performance, which can affect the stack efficiency, stack capacity, or both. Increased pressure drops through various flow passages can also decrease the fine balance developed in fully integrated systems.

- Contaminants are the major concern when operating on alternative gaseous fuels. If any additional sulfur and other components (e.g., chlorides) can be removed prior to entering the fuel processing catalyst, there should be no performance or life impact. If not, the compounds can cause decreased fuel processor catalyst life and potentially impact stack life.

Availability

Although fuel cell systems are perceived as low maintenance devices, their technical immaturity and market entry status cause concern in DG applications. Close attention has been given to the availability of the initial fleet of over 200 commercial PAFC fuel cell units. In a recent 12-month period, the fleet of units in North America achieved 89% availability, with 94% during the last 30 days of the period. In premium power applications, 100% customer power availability, and 96.3% fleet availability has been reported during the same period.[24] This performance is a preliminary indicator that fuel cells can provide high levels of availability, even in high-load factor applications.

The use of multiple units at a site can further increase the availability of the overall facility. Analysis conducted during the fuel cell field demonstration programs of the 1980s indicated that three to five units sized to 120% of application load, operating in parallel, could provide 99.99%-plus availability under typical commercial building load profile characteristics.

[24] According to manufacturer United Technology Corporation (*www.UTCFuelCells.com*, 3/28/02).

Emissions

Fuel cell systems produce few emissions since the primary power generation process does not involve combustion. In fact, the fuel processing subsystem is the only significant source of emissions. The anode-off gas that typically consists of 8 to 15% hydrogen is combusted in a catalytic or surface burner element to provide heat to the reforming process. The temperature of this lean combustion can be maintained at less than 1,800°F, which also prevents the formation of oxides of nitrogen (NO_x) but is sufficiently high to ensure oxidation of carbon monoxide (CO) and volatile organic compounds (VOCs – unburned, non-methane hydrocarbons). Typically, an absorbed bed before the fuel processor removes and eliminates other pollutants such as oxides of sulfur (SO_x).

Nitrogen Oxides (NO_x)
Three mechanisms form NO_x: thermal NO_x, prompt NO_x, and fuel-bound NO_x. Thermal NO_x is the fixation of atmospheric oxygen and nitrogen, which occurs at high combustion temperatures. Flame temperature and residence time are the primary variables that affect thermal NO_x levels. The rate of thermal NO_x formation increases rapidly with flame temperature. Early reactions of nitrogen modules in the combustion air and hydrocarbon radicals from the fuel form prompt NO_x. It forms within the flame and typically is on the order of 1 ppm at 15% O_2, and is usually much smaller than the thermal NO_x formation. Fuel-bound NO_x forms when the fuel contains nitrogen as part of the hydrocarbon structure. Natural gas has negligible chemically bound fuel nitrogen. Fuel-bound NO_x can be at significant levels with liquid fuels.

Carbon Monoxide (CO)
CO and VOCs both result from incomplete combustion. CO emissions result when there is inadequate oxygen or insufficient residence time at high temperature. Cooling at the combustion chamber walls and reaction quenching in the exhaust process also contribute to incomplete combustion and increased CO emissions. Excessively lean conditions can lead to incomplete and unstable combustion and high CO levels.

Unburned Hydrocarbons
Volatile hydrocarbons, also called volatile organic compounds (VOCs), can encompass a wide range of compounds, some of which are hazardous air pollutants. When some portion of the fuel remains unburned or just partially burned these compounds discharge into the atmosphere. Some organics are carried over as unreacted trace constituents of the fuel, while others may be pyrolysis products of the heavier hydrocarbons in the gas. Volatile hydrocarbon emissions from reciprocating engines are normally reported as non-methane hydrocarbons (NMHCs). Methane is not a significant precursor to ozone creation and smog formation and is not currently regulated. Methane is a green house gas and may come under future regulations.

Carbon Dioxide (CO_2)
While not considered a pollutant in the ordinary sense of directly affecting health, emissions of carbon dioxide (CO_2) are of concern due to its contribution to global warming. Atmospheric warming occurs since solar radiation readily penetrates to the surface of the planet but infrared (thermal) radiation from the surface is absorbed by the CO_2 (and other polyatomic gases such as methane, unburned hydrocarbons, refrigerants, and volatile chemicals) in the atmosphere, with

resultant increase in temperature of the atmosphere. The amount of CO_2 emitted is a function of both fuel carbon content and system efficiency. The fuel carbon content of natural gas is 34 lbs carbon/MMBtu; oil is 48 lbs carbon/MMBtu; and (ash-free) coal is 66 lbs carbon/MMBtu.

Fuel Cell Emissions Characteristics

Table 6 illustrates the emission characteristics of fuel cell system. Fuel cell systems do not require any emissions control devices to meet current and projected regulations.

Table 6. Estimated Fuel Cell Emission Characteristics without Additional Controls*

Emissions Analysis[25]	**System 1**	**System 2**	**System 3**	**System 4**	**System 5**	**System 6**
Electricity Capacity (kW)	200	10	200	250	2000	100
Electrical Efficiency (HHV)	36%	30%	35%	43%	46%	45%
Fuel Cell Type	PAFC	PEM	PEM	MCFC	MCFC	SOFC
Emissions[26]						
NO_x (ppmv @ 15% O_2)	1.0	1.8	1.8	2.0	2.0	2.0
NO_x (lb/MWh)	0.03	0.06	0.06	0.06	0.05	0.05
CO (ppmv @ 15% O_2)	2.0	2.8	2.8	2.0	2.0	2.0
CO (lb/MWh)	0.05	0.07	0.07	0.04	0.04	0.04
VOC (ppmv @ 15% O_2)	0.7	0.4	0.4	0.5	1.0	1.0
VOC (lb/MWh)	0.01	0.01	0.01	0.01	0.01	0.01
CO_2 (lb/MWh)	1,135	1,360	1,170	950	890	910
Carbon (lb/MWh)	310	370	315	260	240	245

** Electric only, for typical systems under development in 2002. Estimates are based on fuel cell system developers' goals and prototype characteristics. All estimates are for emissions without after-treatment and are adjusted to 15% O_2.*
Source: Energy Nexus Group.

[25] Emissions estimates are based on best available data from manufacturers and customer data. Emission expressed in lb/MWh are for electric only performance and do not credit emissions for CHP operations. Typically CHP emissions are calculated by Emissions = (lb emissions/(MWh of Elec generated + (MWh of Heat Recovered/80% Boiler eff)*(ratio of Boiler Regulations/Electric Regulations both in lb/MWh equivalent))) and then compared to the Electric Only Regulations.

[26] Conversion from volumetric emission rate (ppmv at 15% O_2) to output based rate (lbs/MWh) for NO_x, CO, and VOC are based on the following conversion multipliers: (0.01418 lb/MWh per ppm NOx) times (System Elec Efficiency LHV); (0.00977 lb/MWh per ppm of CO) times (System Elec Efficiency, LHV); and (0.00593 lb/MWh per ppm of VOC) times (System Elec Efficiency, LHV) respectively.